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The Transmission of Heat in Boilers

CHAPTER II

Condensation of Steam and Rate of Heat Transmission in Surface Condensers and Coolers

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HEAT TRANSMISSION IN BOILERS
CONDENSERS AND EVAPORATORS

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PREFACE

THE experimental data and results given in Chapter IV of the volume on *Heat Transmission by Radiation, Conduction, and Convection* were mostly obtained from apparatus of small dimensions. Although these results illustrated the laws of the convection of heat and their relation to the resistance offered to the flow of fluids, they could not fairly be considered to be directly applicable to large boilers or heaters without further investigation. The first chapter of the present volume is, therefore, devoted to the consideration of experiments on boilers and to comparisons with the results obtained from the small apparatus previously mentioned, as well as to examples of the application of the laws of heat transmission to design. Although a large number of isolated tests have been made on boilers of various types, relatively there are but few results available of an authentic character where the boiler has been tested at various rates of evaporation and where a fair analysis of the losses could also be made.

The problems associated with the transmission of heat in surface condensers and similar plant are quite as complicated as in boilers, but for different reasons. Here again in Chapter II only authentic experiments have been discussed, and an examination of the results shows how variable may be the rate of heat transmission from the steam to the water when condensation takes place in the presence of air. The transmission of heat in evaporative condensers and in cooling towers and ponds is similarly complicated. The transmission of heat in evaporators is more or less dependent upon the same factors as in surface condensers, but is further complicated by the conditions associated with the circulation and evaporation of the boiling liquid or solution undergoing concentration, particularly when the evaporator is of the multiple-effect type.

No claim is made that the various experimental data lead to absolutely conclusive results under all conditions. Much

research work is still necessary, and a careful study of the results presented will indicate what problems still await a more complete solution. For this reason the various experimental apparatus and results are described and discussed in some detail, and in plotting results all the experimental points are shown in the various graphs whenever possible, even at the risk of making some of the graphs appear somewhat involved. This was felt to be necessary to give the reader an opportunity of judging the relative importance of each set of experiments, and the plotted points not only show the variations from the mean, but they also indicate the number of experiments included in each series.

It would have been outside the scope of the present work to enter into a description of the various types of boilers commonly used in practice. Several standard text-books are available, giving detailed descriptions of these boilers. Similarly no attempt has been made to describe the several types of condensers, air-pumps, or cooling towers used in present-day practice. This has been reserved or treatment in a separate work, entitled *Condensers, Air-Pumps, and Evaporators.* Only sufficient description or illustration of evaporators is given in the present volume to enable the reader to understand the experimental apparatus and the results obtained, and a detailed description of evaporators and the principles involved has also been reserved for separate treatment in the above-mentioned book.

Acknowledgment is due to the Institution of Civil Engineers, Institution of Mechanical Engineers, Institution of Naval Architects, and the Institution of Engineers and Shipbuilders in Scotland for information and illustrations from their several *Proceedings*, as well as to several American and other publications referred to in the text. As it is hardly possible to mention in the text all the sources of information in detail, the author here takes the opportunity of expressing his indebtedness to Professor A. L. Mellanby for much valuable data and information concerning the transmission of heat in boilers and condensers.

R. ROYDS.

October, 1920.

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CHAPTER I

THE TRANSMISSION OF HEAT IN BOILERS

CHAPTER II

CONDENSATION OF STEAM AND RATE OF HEAT TRANSMISSION IN SURFACE CONDENSERS AND COOLERS

HEAT TRANSMISSION
IN BOILERS, CONDENSERS, AND EVAPORATORS

CHAPTER I

THE TRANSMISSION OF HEAT IN BOILERS

ALTHOUGH the experiments and their results discussed in Chapter IV of *Heat Transmission by Radiation, Conduction, and Convection* are extremely interesting and instructive, it might legitimately be argued that the results of experiments made on small plant under laboratory conditions do not necessarily apply to boiler conditions in practice. Therefore it is proposed to describe and to discuss certain experiments made on full-sized boilers, and then to compare the results, as far as is possible, with those obtained on small apparatus. Before doing this, however, it would perhaps be well to consider briefly some of the principal difficulties associated with a series of experiments on full-sized boilers. Summarised, these difficulties might be stated as follows :—

1. The heavy expenses of operation on large boilers for experiments over long periods, particularly when special precautions are taken to ensure reasonable accuracy, and when an attempt is made to measure all the various losses of heat.
2. The great difficulties associated with the accurate measurement of the temperatures of a moving gas in large boilers.
3. Particularly with coal fuel, it is almost impossible always to control the combustion and the air supply to the furnace to accord with desired conditions, and with high rates of

combustion a large and uncertain amount of fuel may be carried through the boiler and emitted as sparks and unburnt hydrocarbons. This causes erratic results, even when the analyses of the fuel and of the flue gases are reasonably accurate.

It would hardly be necessary to enlarge upon the expensive nature of boiler experiments. With regard to the measurement of gas temperatures, it might be said that, even in the case of large boilers, it is the common practice to measure the flue gas temperature by a single thermometer, having the bulb well inserted into the current of gas. Except when the boiler works lightly, such a position is probably sufficient to get an average temperature reading, seeing that the flow of gases is usually more or less turbulent. Quite apart from thermometer errors, however, there is one source of error which is nearly always neglected, and that is, the influence of radiation from or to the thermometer bulb or cover. Usually the thermometer, whatever its type, is placed where it is more or less subject to the radiations from the comparatively cold boiler tubes or drums, which causes it to read below the true temperature of the gas. Some examples of this are given on p. 18 of *Heat Transmission by Radiation, Conduction, and Convection.*

In the following discussion attention is directed only to boiler tests of an authoritative character made at various rates of working, in which attempts have been made to measure the furnace temperatures by means of modern pyrometers, and which allow of, at least, an approximate analysis of the various losses of heat. It is also proposed to compare the rates of heat transmission in large boilers with the results obtained, and discussed previously, on small plant working under laboratory conditions, besides which, boiler efficiencies are considered as well as the various losses of heat at different rates of working.

Nicolson's Boiler Experiments.—In order to test thoroughly his ideas of the laws of heat transmission based on Reynolds' work, and as verified by experiments on small laboratory plant, the late Professor J. T. Nicolson designed, and had constructed, various arrangements applied to an ordinary Cornish boiler. The results of some of his experiments were mentioned briefly by him in a lecture before the Junior

Institution of Engineers in 1908, but the apparatus and the results of some of his experiments are given more fully in his paper on "Boiler Economics and the Use of High Gas Speeds."* The experimental arrangements adopted were a complete departure from ordinary practice. Unfortunately, the tests made were not so numerous as one could wish, partly on account of their expensive nature and partly because failing health prevented Professor Nicolson from following up the work as he would have liked. The objects of the construction, running, and testing of this large experimental boiler plant were stated by him as follows :—

1. To show that a greatly increased rate of evaporation per unit of heating surface could be attained by a mere increase of the gas speed.

2. That this increased evaporative power could be got without any sacrifice of evaporative or thermal efficiency.

3. To prove, by continuous working, that the narrow flues which the use of high gas speed entails do not choke up with tar or coal dust, and do not corrode at the outlet end of the flues.

4. To show that by the use of high speed flow of the feed water in small bore tubes, the deposit of sediment is avoided, and that corrosion due to pitting cannot take place.

5. To verify Osborne Reynolds' law of heat transmission.

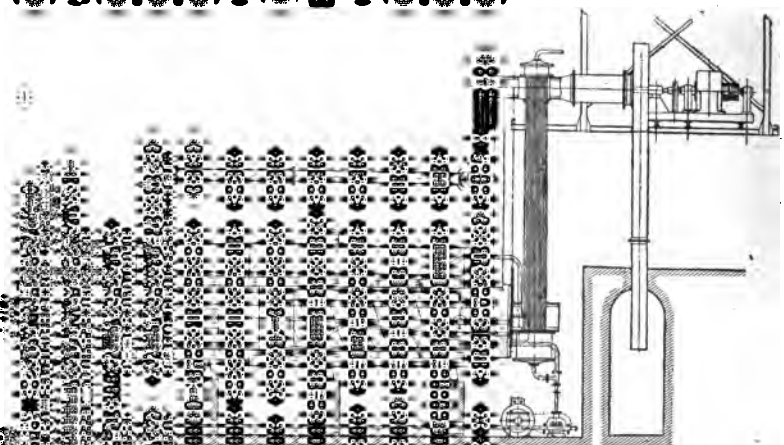
It will first be necessary to describe the arrangements and construction of the experimental plant before going on to discuss the results obtained. A Cornish boiler was placed at Professor Nicolson's disposal by Messrs. Joseph Adamson and Company, Ltd., of Hyde. This boiler was 6 ft. 6 in. mean inside diameter, and 24 ft. long. It had one internal flue of Adamson ring construction, 3 ft. 5 in. inside diameter. The fire grate area, exclusive of dead plate, was 18·1 sq. ft.

The First Arrangement.—The first arrangement tried is illustrated in Fig. 1. It should be understood at the outset that this was *not* a proposal to modify Cornish or similar boilers to the design illustrated, but to obtain data from which it was hoped high-speed boilers could be readily designed, and for the objects mentioned above. The following description

* *Trans. Inst. Engs. and Shipbds. in Scotland,* Vol. LIV, 1910–11.

[illegible] to p. 13 are largely [illegible] paper.

[illegible] here was placed within [illegible] ular water-drum which [illegible] g only a narrow space [illegible] oducts of combustion. [illegible] he circulation of water [illegible]; and through these [illegible] r itself was forced by [illegible] g a capacity of about [illegible] red at the front end of



[illegible] son's boiler experiments.

[illegible] o pass in close contact [illegible] by means of a casting [illegible] a very narrow opening [illegible] er, or mixed water and [illegible] ular space, $\frac{1}{2}$ in. wide, [illegible] ple threaded helix of [illegible] back end of the water- [illegible] to the boiler itself. The [illegible] conomiser) also passed [illegible] with the discharge from [illegible] rum.

[illegible] 3 vertical tubes $\frac{7}{8}$ in.

outside diameter, $\frac{3}{4}$-in. bore, 14-ft. long, and $1\frac{1}{8}$-in. pitch, placed inside of a 16-in. sheet steel pipe. Each tube had a square iron rod of $\frac{1}{2}$-in. side inserted within it and passing through its whole length. Very narrow channels were thus formed for the fresh feed to travel through, so that its speed of circulation might be sufficiently high even when the feed pump was running at a slow speed. In this first arrangement the feed entered at the top, flowed downwards into the bottom header, and so via the water-drum into the boiler. The gases, after being drawn through the narrow flue round the water drum into a dust depositing box at the back, were led round and about the economiser tubes in an upward direction within the 16-in. pipe, which contained the 163 ($\frac{7}{8}$-in.) tubes, and which formed a smoke-flue leading to the fan. The fan was 42 in. diameter over the tips, and was capable of producing a vacuum of 16-in. water gauge when running at 1640 revs. per min., and discharging 10,000 cub. ft. of air per min. at a temperature of 300° F. It was driven by a 50 B.H.P. electric motor.

Several runs were made with the plant thus arranged, the first being on July 13th, 1908. The results obtained on this date were as follows :—

Coal fired per hour (Ripley screenings), 1200 lb. ; being at the rate of 63 lb. per hour per square foot of grate.

Temperature of waste gases to fan, 340° F. to 400° F.

,, ,, entering feed, 80° F.

,, ,, water at bottom header of economiser 338° F.

Draught at fan suction, 21-in. water gauge.

,, ,, back of water drum, $8\frac{1}{4}$-in. water gauge.

It had been hoped that the high speed of the combined feed and auxiliary water circulation in the spirals of the water-drum would sweep along the steam formed therein and prevent it from accumulating in the upper part of the annular water-drum. This hope was not realised, however, and a large amount of steam collected, more especially at the front end of the drum. The outer shell, consequently, got red hot, bulged, and leaked.

It had also been expected that any steam that might be

generated in the economiser tubes would, owing to the high speed of flow of the forced feed circulation, be carried down with the water into the lower header. So long as the feed pump was kept going at a good speed this appeared to take place ; but at slow speeds of the pump the feed water remained stagnant in some of the feed pipes (instead of flowing steadily downwards in all) ; steam was generated in these, and rose into the top or entering feed header, and the proper operation of the counter-current principle was interfered with. Hence the temperature of the waste gases to the fan did not fall below that corresponding to the steam pressure (about 340° F.) ; and this inverted method of supplying the feed water, which had the advantage of delivering the gas to the fan suction by a very short duct (as the fan plant had to be suspended from the roof principals) had to be abandoned.

The Second Arrangement.—The economiser was therefore turned downside up, so that the feed entered at the bottom and any steam formed rose to the top. At the same time, new gas ducts were provided so that the furnace products entered at the top and left at the bottom of the economiser. The water-drum in the flue was taken out, repaired, and replaced as before, but with a special steam escape pipe from the front top end of the water space, Fig. 1.

This arrangement was tried on September 24th, 1908, with the following results :—

Coal fired per hour (Ripley screenings), 1300 lb., equivalent to 68 lb. per square foot of grate per hour.

Temperature of gases leaving water-drum flue, 820° F.
,, ,, gases leaving economiser, 170° F.
,, ,, feed entering economiser, 71° F.
,, ,, feed leaving economiser, 279° F.
,, corresponding to a boiler pressure of 60 lb. per sq. in., 307° F.

Draught at fan suction	$23\frac{1}{2}$ in.
,, ,, bottom of economiser .	23 ,,
,, ,, top of economiser . .	7 ,,
,, ,, back of water-drum . .	$6\frac{1}{2}$,,

The remarkable rate of heat transmission from gas to water in way of the "water-drum flue" may here be pointed out. It was not then possible to measure the water actually evapor-

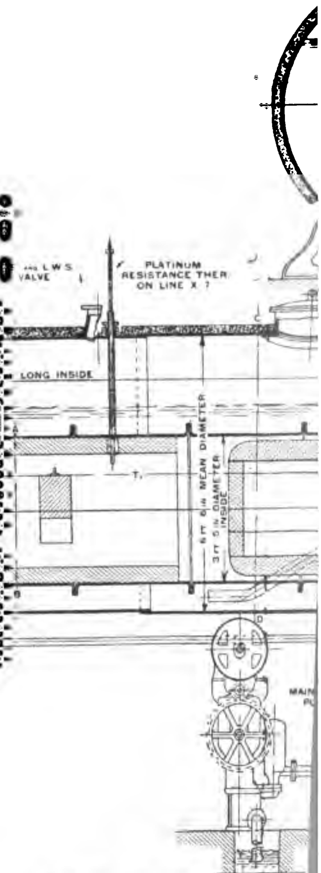


Fig. 2.—Third arrangement

ated; but the amount may be estimated within certain limits of error by the fall of temperature of the gases between furnace and economiser. Assuming a combustion chamber temperature of 3000° F. (afterwards determined by observations at similar rates of firing by Mr. Michael Longridge), it is clear that each pound of the products of combustion of assumed thermal capacity 0·25 B.Th.U. gave up $\cdot 25 \times (3000-832)=542$ B.Th.U. while passing from the brick-lined combustion chamber to the dust box. Taking only 14 lb. of air supplied per lb. of coal $(1300 \times 15)=19{,}500$ lb. of gas flowed through the drum flue per hour; and gave up $19{,}500 \times 542=10{,}580{,}000$ B.Th.U. to the outer surface of the water-drum and the inner surface of the furnace flue. The combined area of these two heating surfaces was 228 sq. ft. Thus the average heat transmission must have been of the order of $\frac{10{,}580{,}000}{228}=46{,}400$ B.Th.U. per sq. ft. per hour. This corresponds to an evaporation of about 48 standard units per sq. ft. per hour.

The steam escape pipe provided to permit of the discharge of the very large quantity of steam formed in the drum at its front end was not able, however, to prevent steam from accumulating therein. The drum consequently again got overheated, and had to be abandoned. The results were, notwithstanding, of so encouraging a character that it was felt that some further attempt should be made to make use of the enormous rates of heat transference which the high gas speed in the narrow flues had shown to be possible.

The Third Arrangement.—It was accordingly determined to remove the water-drum from the boiler flue, and substitute for it a brick plug 38 in. diameter and 10 ft. long, leaving a space of $1\frac{1}{2}$-in. all round between it and the (41-in.) flue, Fig. 2. As the gas temperature was not expected to fall below 1400° F. with this arrangement, it was further decided to fit a vertical small tube "evaporator" between the back of the plug and the top of the economiser. This evaporator was of the same design as the economiser, except that it consisted only of 102 tubes $\frac{5}{8}$ in. outside diameter, $\frac{3}{8}$-in. bore, and 12-ft. long; the central portion of the 16-in. containing pipe being filled up with a 6-in. pipe to which the gas had no access.

It was arranged that the feed upon leaving the economiser should go either directly into the boiler, or go there after mixing with the circulating water drawn by a rotary pump from the boiler and forced through the tubes of $\frac{3}{8}$-in. bore of the evaporator, and again into the boiler, so as to accelerate the circulation in the same. The arrangement is shown in Fig. 2.

The rearranged plant was first tried on January 8th, 1909, and the following results obtained :—

Coal fired 840 lbs. per hour (Ripley screenings), or at the rate of 44·2 lbs. per sq. ft. of grate per hour.

Temperature of gases in combustion chamber, 3000° F.*
„ „ gases leaving brick plug, 1200° F.
„ „ gases leaving evaporator, 620° F.
„ „ gases leaving economiser, 140° F.
„ „ feed entering economiser, 70° F.
„ „ feed leaving economiser, 270° to 340° F.
„ corresponding to boiler pressure of 120 lb. per sq. in., 340° F.

It will be observed that the waste gas temperature fell in this experiment to within 70° F. of that of the entering feed.

The transmission through the heating surface in way of the plug flue was in this instance roughly $840 \times 15 \times \cdot 25 (3000-1200) = 5{,}670{,}000$ B.Th.U. per hour; or at the rate of $\frac{5{,}670{,}000}{118} = 48{,}000$ B.Th.U. per hour per sq. ft. of heating surface. No doubt some of this heat transmission was due to radiation from the brickwork. The heat transmission in the economiser can similarly be estimated. Here the mean temperature difference between gas and water was much less than in the plug flue, being about 200° F. only, instead of 1600° F. The heat transmitted was about $12{,}600 \times \cdot 24(620-140) = 1{,}450{,}000$ B.Th.U. per hour. The heating surface being 521 sq. ft., the amount per square foot of tube surface was $\frac{1{,}450{,}000}{521} = 2785$ B.Th.U. per hour. This is as much as some boilers give as an average for the whole (excluding economiser) of their heating surface. The effect of gas speed in promoting rapidity of heat transference

* Estimated.

was in this way definitely shown; and there seemed to be a possibility by its use of greatly reducing the ratio of heating to grate surface below the usual value, without at the same time causing the diminution of efficiency which has always hitherto been associated with forced rates of combustion and evaporation in steam boilers of every type.

It was accordingly decided to keep the boiler under steam for several months, and to make observations of the various temperatures both of the gases and feed water, and of the draught vacua at several points in the flues. It was also desirable to weigh the coal and measure the feed, and for this purpose tanks and a weigh scale were, after a while, made available.

The principal object in continuously running this experimental plant was, of course, to ascertain on the one hand whether the narrow gas flues would become blocked up with coal dust on the outside; and, on the other, to observe what would become of the sediment and gases contained in the feed when set free in the narrow water channels of the economiser and evaporator.

The Gas Flues.—The long economiser tubes of $\frac{7}{8}$ in. diameter and pitched $1\frac{1}{8}$ in. from centre to centre, had been somewhat improperly handled, both by unequal expansion when first used in the inverted position and (probably) when being taken out, turned end for end, and replaced before the trial of September 24th, 1908. In some places they were too widely spaced, and in others they were standing packed together in groups.

The natural consequence was that coal dust and soot began to bridge across from tube to tube, and a vacuum of even 27 in. of water produced by the fan was latterly found to be insufficient to burn the required quantity of coal.

It was therefore decided to re-tube the economiser with 90 tubes instead of the 163 tubes, at a wider pitch, and with an iron pipe 6 in. diameter forming a central core to restrict the area of the gas passage. With such more widely pitched tubes of the same total length (the old tubes were reinserted) so low a waste gas temperature as 140° F. could no longer be expected; but calculation foretold that it might be expected

to fall to about 220° F. On March 22nd, 1909, this re-tubing was completed, and on that date a trial under the new conditions was made.

It was found that the waste gases were reduced to 190° F., notwithstanding the wider pitching of the tubes. This low gas-outlet temperature was, however, only maintained while the tubes were perfectly clean. It was found that, after a few days, a thin coating of coal dust had adhered to the tubes, and the average value of the outlet temperature had risen to 240° F.

On the trials made six months afterwards, in October, 1909, by Mr. Michael Longridge, after fifty days of running of the plant at draughts of from 15 to 20 in., the waste gas temperatures were, however, found to be of substantially the same value. Thus the soot film was found to have attained, as a permanent regime, under the scouring action of the gases, a thickness sufficient to prevent all corrosion of the tubes, but not sufficient to prevent the gases from transferring their heat to the metal at a much greater rate than that commonly found in steam boilers.

The Water Channels.—It was also found that in the very narrow water spaces of the economiser (viz. those left by the $\frac{1}{2}$-in. square rod placed inside of the $\frac{3}{4}$-in. bore tubes) there was no deposit either upon the rods or the tubes. There was also practically no corrosion, although Mr. Longridge found "slight but unmistakable pitting by oxygen" on the upper ends of the rods when he examined the economiser in October, 1909. This pitting was, however, probably due to the water standing in the tubes at nights when the fire was banked and the feed pump shut off. The upper parts of the evaporator tubes (of $\frac{5}{8}$-in. outside diameter and $\frac{3}{8}$-in. bore) above the level of the boiler water were also free of sediment, notwithstanding the fact that the dirty water* from the economiser had been passed through the evaporator on its way to the boiler during the whole fifty or sixty days the plant had been at work.

Below the boiler water level, however, the evaporator tubes had become gradually blocked up inside with a white sediment which was very hard and adherent for the first foot or two

* The feed water was untreated canal water.

from the bottom, but soft enough to be pushed out by a ¼-in. rod between that and the water level.

In order to understand how this occurred the arrangement of the plant must be looked at, as shown in Fig. 2.

When the evaporator was fitted (in order to provide the additional surface necessitated by the removal of the " water drum ") it was, in order to save expense, simply passed through the old " ash box " in way of the old bye-pass ; that pipe being, in fact, itself made use of as part of the containing pipe for the new evaporator. The new tubes passed vertically completely through the "ash box," and took their water supply from a water-header placed below the same.

It resulted from this that during the night, when no current was available to drive the circulating pump, and the feed donkey was also shut down, there was no circulation whatever through the evaporator tubes except that due to the gravity displacement by the boiler water of so much of the water in the tubes as was turned into steam and passed away into the boiler steam space. From 700 to 1000 lb. of coal were burnt during the fifteen hours of banked firing which took place on each of the fifty or sixty days the plant worked, and the only way by which the gases so produced could get to the chimney was by flowing around and amongst the lower ends of the evaporator tubes on its way to the bye-pass duct.

The result was the choking with sediment *up to the level of the boiler water,* described above.

The extent to which this choking of the tubes in their lower parts had proceeded was not realised until shortly before the official trials by Mr. Longridge were to take place.

The circulating pump, which drew from the boiler and passed water upwards through the evaporator tubes (along with the fresh feed from the economiser) began (in September, 1909) to take so much current through its armature that it was constantly stopping owing to the fuse blowing ; and when the plant was opened up to find the cause of this great hydraulic resistance, it was found, as already mentioned, that quite a number of the tubes were completely choked, that many of them had only a fine hole through their centres, so that, whilst enough remained clear to allow of the feed pump passing the

relatively small quantity of fresh feed into the boiler through the evaporator without difficulty, the rotary pump was opposed by a greatly restricted area for the flow of its 4000 gallons an hour. Professor Nicolson reported this state of things to Mr. Longridge, and invited him to make an inspection of the plant, so that he might satisfy himself that the cause of the choking had been the boiling of the water in the pipes during the night when no positive circulation was possible.

This inspection took place on 22nd September, 1909. Overheating and bending of a few of the evaporator tubes was then found, in the case of those tubes only which had been completely choked with sediment and which had been exposed for some days, or even weeks, to the contact not only of the moderately hot gases at nights, but also of the products at 1200° F. to 1500° F. which passed over them at high speed during the daytime. Notwithstanding this ill-usage, not a single tube gave way, nor was there any sign of leakage at the tube plates. The results of this experiment seem to point to the feasibility of using small bore tubes with thick walls to withstand the fierce heating action of high-speed gas at high temperatures when a positive circulation at moderate velocity is available.

The plant was laid off for a week or two before Mr. Longridge's trials in order to allow of the removal of as much as possible of the hard scale from the lower ends of the evaporator tubes. No cleaning of any part of the sooty outside surfaces of the tubes of the economiser or evaporator was attempted, or was, indeed, possible.

So far as could be seen there was but little adherent matter on the inside of the furnace flue in way of the brick plug ; but whatever there might be was not scraped off, as it was not believed to be seriously prejudicial to heat transmission. The plant went, therefore, into Mr. Longridge's hands for testing just as it was after fifty-one days of steaming.

Referring again to Figs. 1 and 2 it is seen that the furnace and fire grate were of ordinary construction. Behind the fire bridge, however, a reverberatory chamber was formed of firebrick to promote the complete combustion of the hydrocarbon gases given off by the fuel. Professor Nicolson found

that little or no smoke was produced even at the heavy rates of firing adopted in his experiments, and he considered such an arrangement to be far preferable to the more common method of promoting combustion by screening the heating surface over the grate by means of firebrick, since the latter arrangement cuts off most of the radiant heat from the fire and thereby screens one of the most effective portions of the heating surface in a boiler. Professor Nicolson's arrangement allowed the radiant heat from the fire to pass into the heating surface, and only screened a later and relatively less effective portion of the heating surface from contact with the gases.

Mr. Michael Longridge made five complete tests on the boiler arranged as shown in Fig. 2. For the full details of Mr. Longridge's report reference might be made to the 1909 report of the British Engine, Boiler, and Electrical Insurance Company or to Professor Nicolson's paper on "Boiler Economics and the Use of High Gas Speeds." It is sufficient for our present purpose to quote only a few of the tabulated data and the results obtained. These are given in Table 1.

TABLE 1

Grate surface (exclusive of dead plate)	18·1 sq. ft.
Heating surface :—	
Above fire-grate	41 sq. ft.
Combustion chamber not counted as heating surface	0 ,, ,,
Internal flue beyond combustion chamber	130 ,, ,,
	171
Evaporator	192 ,, ,,
Economiser	293 ,, ,,
Total	656 ,, ,,
Sectional Area for Gas-flow :—	
Annular passage through back part of internal flue . about	1·7 sq. ft.
Through evaporator	·985 ,, ,,
Through economiser	·823 ,, ,,
Sectional Area for Water-flow :—	
Through evaporator tubes	·0782 sq. ft.
Through economiser tubes	·0356 ,, ,,
Hydraulic Mean Depth on Gas Side of Tubes, m, :—	
Evaporator: For heat-flow	·709 in.
Evaporator: For resistance to gas-flow	·527 ,,
Economiser: For heat-flow	·479 ,,
Economiser: For resistance to gas-flow	·375 ,,

Values of $\frac{l}{m}$:—

Evaporator ($l = 11$ ft.) .	For heat-flow . . .	186·5
	For resistance to gas-flow	251
Economiser ($l = 13·75$ ft.) .	For heat-flow . . .	345
	For resistance to gas-flow	441

	Number of test . . .	1	2	3	4	5	6
	Date of test . October, 1909	12	13	14	19	20	21
12	Duration of test, . hours	6·666	9·00	9·834	9·085	6·05	6·65
	Fuel.						
13	Short description . . .	Manvers Main.			Welsh.		
14	Weight fired per hour . lbs.	685	688	413	478	740	737
14a	Weight dried fuel fired per hour . . . . . lbs.	662 6	665 5	399·5	467·5	724·0	721·0
47	Weight dried fuel fired per hour per sq. ft. grate lbs.	36·6	36 8	22·1	25·8	40·0	39·9
17	Calorific value of dried fuel, B.Th.U./lb.	13,166	13,166	13,166	14,929	14,929	14,929
	Average Gas Temperatures.						
22	Leaving flues, T_4 . . °F.	288	279	241	253	291	294
22a	Entering economiser T_3 °F.	797	814	677	689	870	871
22b	Entering evaporator T_2 °F.	1252	1322	1184	1138	1360	1418
22c	In combustion chamber T_1 °F.	2162	3000	2000	1945	3000	3000
23b	Weight of gases per lb. dried fuel (from analysis of flue gases) lbs.	—	15·39	19·52	20·32	14·13	14·66
	Draught.						
27	Vacuum in furnace, in. water	—	0·35	0·20	0·18	0·28	0·28
27b	Vacuum in dust-box, in. water	2 75	2·41	1·49	1·80	2·65	2·85
28	Vacuum in flue at top of economiser . . in. water	8·60	7·56	4 80	5·60	8·15	8·68
29	Vacuum in flue leaving economiser . . in. water	19·75	17·20	11·30	12 75	18·00	19·00
30	B.H.P. given by motor driving fan	39 8	34·1	20·2	26·4	37·1	36·9
	Feed Water.						
31	Total per hour . . lbs.	5890	5810	3625	4730	6980	7040
31a	Equivalent from and at 212° F. . . . . lbs.	6986	6903	4307	5606	8265	8372
	Temperature of Feed.						
32	To economiser, t_4 . . °F.	74	73	72	75	77	71
32a	Leaving economiser, t_3 . °F.	284	280	289	291	288	275
32b	Entering evaporator from boiler t' . . . °F.	350	350	350	350	350	350
32c	Mixtures of t_3 and t_1 entering evaporator, t_2 . . °F.	—	334	340	340	338	342
32d	Leaving evaporator to boiler, t'' . . . . . °F.	350	350	350	350	350	350
33	Steam pressure in boiler, lbs. sq. in. gauge . .	120	120	120	120	120	120
53a	Evaporation from and at 212° F. per sq. ft. of boiler and evaporator surface (363 sq. ft.) . . .	19·25	19·02	11·87	15·44	22·77	23·06

53	Evaporation from and at 212° F. per sq. ft. of total heating surface (656 sq. ft.)	10·65	10·5	6·56	8·53	12·58	12·75
38	Thermal efficiency of whole plant, per cent . . .	77·4	76·8	79·1	77·6	73·8	75·0
38a	Estimated net efficiency of whole plant, per cent. .	66·0	66·8	68·9	69 1	64·4	66·1
44	Heat unaccounted for in heat balance, per cent of heat in fuel	—	15·0	11·3	13·2	16·0	13·3
	Average B.Th.U. transmitted per sq. ft. per hr. = H.						
	Through plug flue . . .	—	33,400	12,200	14,540	32,400	32,200
	Through evaporator tubes .	—	6850	5130	5480	6550	7530
	Through economiser tubes .	—	4720	2890	3480	5070	5200
	Mean temperature difference between Gas and Water (t_m).						
	In plug flue	—	1670	1192	1143	1700	1740
	In evaporator . . .	—	685	541	534	737	761
	In economiser . . .	—	345	263	273	368	379
	Average B.Th.U. transmitted per sq. ft. per sec. per deg. Fah. diff. (h).						
	Through plug flue . . .	—	·00556	·00284	·00353	·00530	·00515
	Through evaporator tubes .	—	·00278	·00263	·00286	·00247	·00275
	Through economiser tubes .	—	·00381	·00306	·00354	·00382	·00382
	**Mean Velocities of Gas, ft. per sec. (v_1)*						
	In plug flue	—	104	64	75	105	110
	In evaporator . . .	—	106	75	90	110	115
	In economiser . . .	—	86	60	74	89	91
	Mean Densities of Gas, lbs. per cub. ft. (ρ_1).						
	In plug flue	—	·0178	·0209	·0216	·0176	·0173
	In evaporator . . .	—	·0279	0304	·0306	·0266	·0267
	In economiser . . .	—	·0432	·0465	·0458	·0419	·0421
	**Values of* $v_1\rho_1 = \frac{w_1}{a_1}$						
	In plug flue	—	1·67	1·27	1·55	1·67	1·72
	In evaporator . . .	—	2·89	2·20	2·68	2·88	2·98
	In economiser . . .	—	3·46	2·65	3·20	3·45	3·57
	**Values of* $\frac{H}{3600\, v_1\rho_1(T-t)} = c.$						
	For plug flue	—	·00333	·00223	·00228	·00317	·00299
	For evaporator . . .	—	·00096	·00120	·00106	·00086	·00092
	For economiser . . .	—	·00110	·00115	.00111	·00116	·00107

* All these values are calculated on the assumption of clean tubes. Actually the tubes were covered with a layer of soot, and therefore all these values are purely nominal.

Calculated Values of M *from equation* (6), *p.* 29 .						
For evaporator . . .	—	252	199	221	281	260
For economiser . . .	—	361	413	427	344	350
Values of $v_2 \rho_2 = \frac{w_2}{a_2}$ for economiser . . .	—	45·4	28·3	37·0	54·5	55
Corresponding values of C_2 from Fig. 81 § . . .	—	·0154	·0140	·0147	·0162	·0162
Assumed value of C_2 for evaporator	—	·03	·03	·03	·03	·03
* *Calculated Values of* c_1 *from equations* (18) or (21), *p.*148 §						
For evaporator . . .	—	·00112	·00142	·00130	·00099	·00109
For economiser . . .	—	·00171	·00170	·00174	·00166	·00161
† *Calculated Values of* f' *from equation* (23), *p.* 85 § (*No allowance for kinetic energy.*)						
For evaporator . . .	—	·0202	·0249	·0194	·0212	·0206
For economiser . . .	—	·0230	·0297	·0222	·0230	·0225
‡ *Calculated Values of* f *from equation* (23), *p.* 85 § (*Allowance made for kinetic energy at inlet.*)						
For evaporator . . .	—	·0157	·0202	·0149	·0167	·0160
For economiser . . .	—	·0201	·0281	·0194	·0200	·0196

The numbers in the left-hand column of the Table 1 refer to the number of the line in the standard form of the Institution of Civil Engineers || for reports of boiler tests. The weight of gases passing through the boiler was calculated from the chemical analyses of the flue gases and the coal. On referring to lines 53 and 53*a* it is seen that the rate of evaporation was much greater than obtains under ordinary boiler conditions, and consequently the rate of combustion on the grate was correspondingly high. It is generally found

* All these values are calculated on the assumption of clean tubes. Actually the tubes were covered with a layer of soot, and therefore all these values are purely nominal.

† These values of f' include the effect of end resistances and also the resistance due to the increase of the kinetic energy of flow at the inlet to the tubes.

‡ These values of f include end resistances other than due to the increase of kinetic energy at the inlet to the tubes.

§ *Heat Transmission by Radiation, Conduction, and Convection.*

|| *Proc. Inst. Civil Engs.*, Vol. CL and CXCV; or *Testing of Motive Power Engines.*

that high rates of combustion cause excessive furnace losses, partly because of incomplete combustion and partly because of the small cinders and coal dust carried through the tubes. Line 44 indicates how large was the heat unaccounted for, the greater portion of which could be ascribed to losses by cinders and coal dust carried through the boiler unburned, but which could be avoided by designing a boiler with grates sufficiently large to permit of a lower rate of combustion. Notwithstanding such losses, line 38 indicates the high thermal efficiency of the boiler under the conditions of working. These values, however, make no allowance for the power absorbed by the fan and circulating pump.

Assuming that an engine driving these would use 20 lb. of steam per brake horse power per hour, and assuming that none of the exhaust steam is utilised for feed heating, the net efficiency is given in line 38*a*. There is no reason, however, why the exhaust steam from the fan engine and the other plant, when steam-driven, should not be utilised for feed heating, and in that case the thermal efficiency on line 38 would then represent practically the net efficiency. It may be fairly claimed, therefore, that a high-speed boiler could be made to give an overall efficiency of perhaps 85% if the furnace losses were practically eliminated by adopting a moderate rate of combustion.

Assuming that, to a first approximation, the heat transmitted per hour per sq. ft. (H) can be represented by

$$H = 3600c \frac{w_1}{a_1} t_m$$

where t_m is the average difference of temperature between the gas and the water, a comparison of the calculated values of c given in the table shows that those derived for the plug flue are considerably higher than for the evaporator and economiser. The reason for this is the large amount of heat radiated by the brickwork plug, which was white hot at the furnace end and more than red hot at the other end.

The calculated rate of heat transmission, in B.Th.U. per sq. ft. per sec. per deg. Fahr. difference are shown plotted in Fig. 23, p. 47. The calculated values of M from equation

5 and of c_1 from equation 6, p. 29 are also given in the Table 1. The values for $C_2 = c_2 \frac{w_2}{a_2}$ were taken from Fig. 81,* p. 180, for the economiser, and for the evaporator C_2 was assumed to be constant at ·03.

From the tabulated values of the gas pressures at the various points in the plant the differences of pressure, $\triangle P$, required to force the gases through the evaporator and the economiser are easily calculated. It will be noted that these differences of pressure not only include those required to overcome the resistance to the flow along the tubes, but also include the drop of pressure required to create the kinetic energy of flow† at the inlet to the tubes, and any end losses at the inlet in creating the energy of flow. The character of the flow of gas at the outlet from each section is such that probably all the kinetic energy of flow at the outlet end was wasted. It is possible to calculate approximately the pressure drop required to create the kinetic energy of flow at the inlet in the manner described on p. 53, and if subtracted from the pressure difference, $\triangle P$, can be taken to represent the pressure drop, δP, required to overcome the resistance to flow along the tubes. Making use of equation 23, p. 85,* and inserting the various values obtained from the experiments, the values of f, the coefficient of resistance, are calculable. The values obtained are tabulated in Table 1, p. 16, the mean for the evaporator being $f = ·0167$, and for the economiser, $f = ·0214$, using the calculated pressure drop δP.

If it be assumed that the same equation 23, p. 85,* could be used by inserting the total pressure drop $\triangle P$, then calculating for the values of f_1 the mean values obtained were, for the evaporator $f_1 = ·0213$, and for the economiser $f_1 = ·0251$. For these conditions Professor Nicolson gave a constant deduced from his previous experiments on this plant equivalent to $f_1 = ·013$.

All these values of f are much higher than those obtained in the author's experiments on the small laboratory plant

* *Heat Transmission by Radiation, Conduction, and Convection.*

† The gases would, of course, have some velocity of flow through the end connections between the different parts of the plant, but the corresponding kinetic energy and the resistance are taken to be practically negligible.

given on pp. 187 and 198,* but are more in keeping, however, with those shown in Fig. 73, p. 170,* in connection with the small boilers with tubes of small bore. It must be observed, however, that in course of time the plant settled down to a permanent deposit of soot on the tubes of the economiser and evaporator and therefore the areas for gas-flow given on p. 13 and the deduced values of $\frac{w_1}{a_1}$ may not be the true values under working conditions. But no doubt one of the factors accounting for the high values of f in Nicolson's experiments was the flow of the gases across the tubes at the entrance and at the exit. In crossing the tubes to get access to the inner rows the gas has to acquire an increase of kinetic energy at each tube row it crosses, which is nearly all lost behind each row, and thus this would entail an increase in the pressure energy corresponding to the losses of kinetic energy. This factor is more fully discussed on p. 54.

Professor Nicolson reported that the outlet temperature of the flue gases from the economiser for continuous running was only some 50° F. or so higher than when the plant was started with clean tubes and with the same rate of evaporation. This would seem to suggest that the values of h, c, and c_1, when calculated on the nominal dimensions in such an installation, are not altered greatly by soot on the tubes, which simply means that any increased resistance to the flow of heat caused by the deposit of soot was practically counterbalanced by the increased values of $\frac{w_1}{a_1}$. It should be noted that the permanent deposit of a thin layer of soot on the tubes may be an advantage in preventing corrosion at the low temperature end of the economiser by condensable gases.

Locomotive Experiments.—An extensive series of tests on different types of locomotives were made at the Louisiana Purchase Exposition, Missouri, in 1904, by a committee of the American Society of Mechanical Engineers and the American Railway Master Mechanics' Association, acting in conjunction with the Pennsylvania Railroad Company. The

* *Heat Transmission by Radiation, Conduction, and Convection.*

locomotives were run on a specially designed dynamometer,* a description of which is given in the excellent report† issued by the above-mentioned company, the report also containing the tabulated results and a discussion of the experiments. For the present purpose it is not necessary to consider the engine part of the locomotives, so that the following description and discussion has reference to the boilers only. Altogether eight locomotives of normal types were tested, some particulars of which are given in Table 2; for a description and detail drawings of each locomotive reference should be made to the above-mentioned report.

TABLE 2

	Number of Locomotive.	Type of Locomotive.	Working Boiler Pressure. Lbs. per sq. in. gauge.	Fire Box. Length Inside. ins.	Fire Box. Width Inside. ins.	Grate Area. sq. ft.	Width Air Spaces between Bars. ins.	Heating Surface of Fire-box (Fire Side). sq. ft.
1	1499	Freight. 2 8 0	about 200	116·1	66·0	49·2	0·69	166·4
2	734	2 8 0	about 200	119.4	40·9	33·8	0·69	218·9‡
3	585	2 8 0 Two-cylinder cross compound.	about 210	92·4	66·6	49·4	0·75	165·7‡
4	929	2 10 2 Four-cylinder compound.	about 215	104·2	77·9	58·4	0·94	216·4
5	2512	Passenger. 4 4 2 De Glehn Four-cylinder compound.	about 215	119·9	39·7	33·4	0·50	177·3
6	535	4 4 2 Vauclain Four-cylinder compound.	about 220	115·4	65·9	48·4	1·06	220·3
7	3000	4 4 2 Four-cylinder compound.	about 220	94·9	75·0	49·9	0·75	151·7‡
8	5266	4 4 2 Two-cylinder simple.	about 200	114 0	68·0	55 5	0·75	156·9

* This testing plant is now installed at the Company's works, Altoona, Pennsylvania.

† Or refer to *The Testing of Motive Power Engines*, p. 111.

‡ Including area of arch tubes.

TUBES.

	Diameter on Fire Side d_1 ins.	Diameter on Water Side d_2 ins.	Length. Between Tube Plates. ins.	Length. Approx. Total Length. ins.	Hydraulic Mean Depth, Fire Side m_1 ins.	$\frac{l}{m_1}$ For Heat-flow.	$\frac{l}{m_1}$ For Resistance to Gas-flow.	Number of Tubes.	Heating Surface of Tubes between Tube Plates, sq. ft. Fire Side.	Heating Surface of Tubes between Tube Plates, sq. ft. Water Side.
1	1·73	2	164·5	166·5	0·432	380	384	373	2316	2677
2	1·76	2	178·9	180·9	0·44	406	411	338	2322	2639
3	1·76	2	190·4	192·4	0·44	433	437	363	2653	3015
4	2·0	2·25	238·5	240·5	0·5	477	481	393	4090	4601
5	*	2·75	176·1	178·1	0·325	543	549	139	2479	1469
6	2·0	2·25	225·1	227·1	0·5	450	454	273	2682	3017
7	1·75	2	191·3	193·3	0·44	438	442	390	2848	3255
8	1·75	2	179·8	181·8	0·44	411	416	315	2162	2471

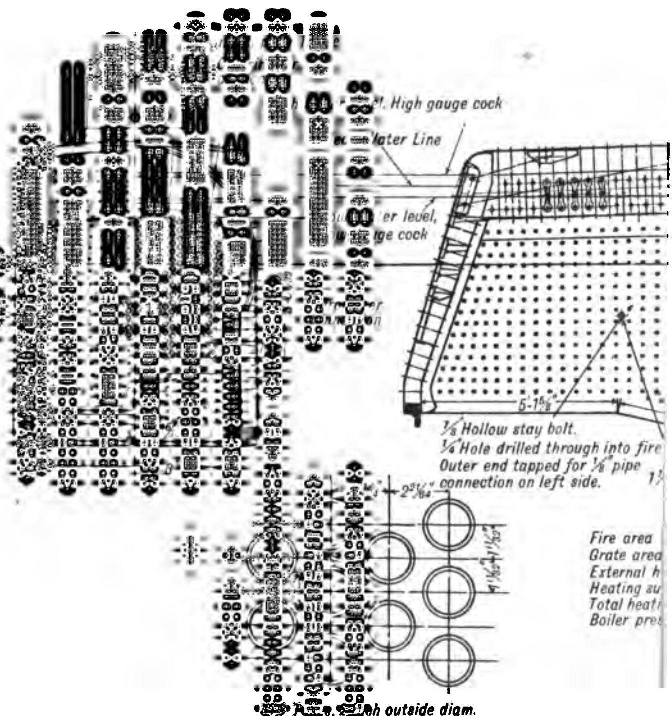
	Total Fire Area through Tubes. sq. ft. a_1	Exhaust Nozzle. Diameter. ins.	Exhaust Nozzle. Area. sq. in.	REMARKS.
1	6·09	5·75	26·0	Owned by Pennsylvania Railroad Company. No fire-brick arch in fire-box ; rocking finger-grates ; no superheater.
2	5·71	5·25	21·6	Owned by Lake Shore and Michigan Southern Railway Company. Fire-brick arch with arch tubes ; rocking grates ; no superheater.
3	6·13	5·5	23·8	Owned by the Michigan Central Railroad Company. Arch tubes ; rocking grates ; no superheater.
4	8·57	6·0	28·3	Owned by the Atchison, Topeka, and Santa Fé Railway System. No fire-brick arch in fire-box ; rocking finger-grates ; no superheater.
5	4·69	variable	max. 43·5 min. 17·5	Owned by the Pennsylvania Railroad Company. Fire-brick arch in fire-box ; stationary wrought-iron grate bars ; no superheater ; Serve tubes.
6	5·96	5·75	26·0	Owned by the Atchison, Topeka, and Santa Fé Railway System. No fire-brick arch in fire-box ; rocking finger-grates ; no superheater.
7	6·51	5·625	24·85	Owned by the New York Central and Hudson River Railroad Company. Arch tubes and fire-brick arch in fire-box ; rocking finger-grates ; no superheater.
8	5·26	5·625	24·85	Owned by the Pennsylvania Railroad Company. No fire-brick arch in fire-box ; rocking finger-grates ; no superheater.

* Serve tubes, ·075 in. thick, having eight ribs, and area of fire-side surface in 1 in. length, 15 sq. ins.

The duration of most of the tests was three hours, such a short period being permissible with the high rates of combustion usual in locomotives. The coal was of good quality but friable, and when the draught was strong, some of the finer particles were drawn through the tubes into the smoke-box. The coal used was carefully weighed out, and samples were taken for analysis. The smoke-box was cleaned out at the beginning and at the end of a test, and the quantity of cinders which had collected were weighed. A special stack was provided over the locomotive with a deflector and receptacle, into which the sparks emitted at the locomotive chimney fell. This receptacle was also cleaned out at the start and at the finish of a test, and the cinders weighed. As the force of the draught on some of the heavier tests carried some of the sparks out at the top of the special stack, it was thought necessary to make some estimate of the amount. These sparks fell on the roof of the building, and for the total period of the tests were estimated to weigh 97,000 lbs. It is interesting to note that the cinders caught in the smoke-box, and those caught in the special stack, altogether amounted to 53,500 lb. for the same period. As about 750 tons* of coal were burned, the total of cinders and sparks thus accounted for amounted to about 10% of the total weight of fuel fired. The feed-water was taken from the town's mains, and treated in a water softener. The amount of water supplied to the injectors was weighed in special tanks, and any water which escaped from the injector overflow pipes was caught and returned to the suction tank. The quality of the steam generated was determined by a throttling calorimeter.

The pyrometers were thermo-couples of platinum, platinum-rhodium wires for measuring the fire-box and smoke-box temperatures. As the one in the fire-box had to be withdrawn frequently to prevent damage during firing, it was found to be necessary to have the wires exposed to the flames in the fire-box, so as to respond quickly to the temperature. Such an exposure, however, made contamination of the wires possible, and they ultimately became brittle, but the wires were annealed and recalibrated when found necessary. It would be seen

* In this case 1 ton = 2000 lbs.



—General positions of draugh

afterwards that the measured fire-box temperatures are most likely below the true temperatures, although the pyrometer was inserted about midway along the fire-box length, and at a height of about 12 in. above the grate.

The pyrometer in the smoke-box remained in position, and except for the possible influence of radiation to and from the relatively cold boiler tube plate, the temperatures could be read fairly accurately. As a check a mercurial thermometer was also inserted in the smoke-box, and the averages of this and the thermo-junction readings have been taken to represent the smoke-box temperatures. The outline illustration in Fig. 3 shows generally the positions of the instruments during the tests.

The calorific value of the coal in each test was determined in a William Thompson type of calorimeter, which was itself standardised by fuels of known calorific value, as determined by several bomb calorimeters.

The ultimate analysis of the dry coal gave :—

Carbon	84·2%	Oxygen	2·94%
Hydrogen	4·28%	Sulphur	0·80%
Nitrogen	1·44%	Ash	6·34%
			100

The approximate weight of gases w_1 passing through the boiler tubes in pounds per second has been calculated by the author for each test from the flue-gas analysis in the following manner, which is given here as an example :—

Test 806. *Locomotive No.* 3000.

Dry coal fired per hour, 3129 lb.

Calorific value of coal (gross), 15,076 B.Th.U. per lb.

Analysis of dry flue gases :—

Carbon dioxide, CO_2	13·63% by volume.
Carbon monoxide, CO	0·10% ,, ,,
Oxygen, O_2	4·37% ,, ,,
Nitrogen, N_2	81·9% ,, ,,

Weight of dry gases formed per pound of coal burned

$$=\frac{11\,CO_2+8\,O_2+7\,(CO+N_2)}{3\,(CO_2+CO)}\times\text{Carbon per lb. coal} \quad . \quad . \quad . \quad (1)$$

where CO_2, CO, O_2, and N_2 represent the proportions by volume of the respective gases. In the above case,

$$\frac{11\times\cdot1363+8\times\cdot0437+7\times\cdot82}{3\,(\cdot1363+\cdot001)}\times\cdot842=15\cdot5 \text{ lb.}$$

Assuming that there would be about 0·5 lb.* of water vapour associated with this gas, partly formed by combustion and partly due to the evaporated moisture from the fuel, and taking 0·24† as the specific heat of the dry gases, then the heat lost in cooling from the smoke-box temperature, 631° F., to atmospheric temperature, 41° F., per pound of coal burned would be :—

$$15\cdot5\times\cdot24\times(631-41)+\cdot5\times1346=2863 \text{ B.Th.U.}$$

The efficiency of the boiler in this test is 64·05%, and the heat lost by carbon burned to CO is about 0·4%. Allowing 3·2% as loss by external radiation, etc., then the total remaining losses are :—

$$100-(64\cdot05+\cdot4+3\cdot2)=32\cdot35\%.$$

If x is the percentage loss by sparks, cinders, unburned hydrocarbons, etc., then, with the gross calorific value of dry coal in this test 15,076 B.Th.U. per pound, and taking the loss x to be equivalent to unburned coal,

$$x+\frac{2863}{15076}(100-x)=32\cdot35$$

$$\text{or } x=16\cdot4\%.$$

Then the percentage loss by the flue gases becomes

$$32\cdot35-16\cdot4=16\% \text{ nearly.}$$

Again, taking the wet gases per pound of coal burned to be $(15\cdot5+\cdot5)=16$ lb., it might be taken that the weight of gases per pound of dry coal fired is as follows :—

$$16\times\left(\frac{100-16\cdot4}{100}\right)=13\cdot38 \text{ lb.}$$

Then, approximate weight of gases per second

$$w_1=13\cdot38\times\frac{3129}{3600}=11\cdot6 \text{ lb.}$$

* With such roughly approximate calculations as are here being considered, this value is sufficiently accurate for the purpose in view.

† Actually the specific heat of such gases increases to some extent with temperature, but the calculation is too approximate to take account of varying specific heat.

and since the total area of section of the tubes available for gas flow a_1 was 6·51 square feet,

Then,
$$\frac{w_1}{a_1} = \frac{11 \cdot 6}{6 \cdot 51} = 1 \cdot 78$$

If it were taken that the coal on the grate radiated according to the Stefan-Boltzmann fourth power law as a black body of flat surface equal to the area of the grate,* that the heating surfaces in the fire-box absorbed the whole of these radiations, and that there could be little or no combustion beyond the fire-box, an approximate estimate of the fire-box temperature might be made by taking it that :—

Heat radiated from fire + Heat given by gases in tubes = Heat received by water in boiler. Thus if T_1 = fire-box temperature, grate area = 49·9 sq. ft., 0·25† = specific heat of gases.

No of Loco.	No. of Test.	Coal fired per hour per sq. ft. of grate lbs.	Calculated furnace temperatures °F.		Measured furnace temperature °F.
			Sp. ht. ·25	Sp ht. ·275	
1499	111	27·8	2340	2270	1480
—	101	86·5	2480	2390	1968

$$\frac{\dfrac{1600 \times 49 \cdot 9}{15076 \times 3129}\left\{\left(\dfrac{T_1+460}{1000}\right)^4 - \left(\dfrac{400+460}{1000}\right)^4\right\} + \dfrac{13 \cdot 38 \times \cdot 25}{15076}}{(T_1 - 631)}$$

$$= \cdot 6405, \text{ the boiler efficiency} \quad \ldots \ldots \quad (2)$$

Inserting $T_1 = 2740°$ F. makes the total ·645, which is near enough for the purpose, seeing that some additional heat would be lost from the boiler surface to the atmosphere. Although this method of estimating furnace temperatures is

* With bituminous coal the flames more or less fill the fire-box, thus extending the radiating surface. Also a small proportion of the total heat would be given to the fire-box surface by the direct contact of some of the gases. Probably, then, the true furnace temperature would be slightly lower than the calculated values.

† The actual mean specific heat of the gases would probably be greater than this value, but the calculation is too rough to make it worth while taking account of variable specific heat. It is seen from the examples given in the following table that the mean specific heat of the gases used in these calculations has only a comparatively small influence on the calculated furnace temperatures.

only approximate it is found to provide a workable basis for boiler calculations.

The fire-box temperature has been calculated by the author in this manner for each test, and as an example of the results

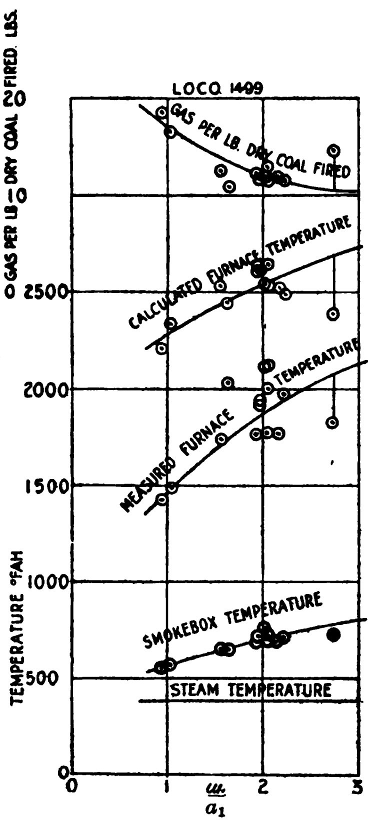


FIG. 4.—Temperatures in locomotive boiler under test.

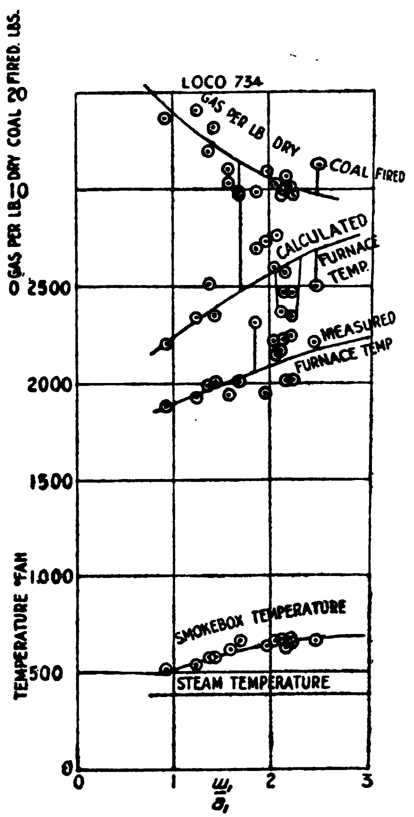


FIG. 5.—Temperatures in locomotive boiler under test.

obtained, Figs. 4 and 5 are reproduced, referring to locomotives Nos. 1499 and 734 respectively, the base line in both cases being $\frac{w_1}{a_1}$. The measured fire-box and smoke-box temperatures are also shown, and it would be seen that the measured fire-box temperatures were certainly too low, even

after making all allowances for the approximate method of calculation.

The large errors involved in measuring gas temperatures, particularly when the thermometer is more or less surrounded by cold surfaces, was discussed on p. 2. For this reason the calculated temperatures are considered to be more nearly correct than the measured values under the conditions of

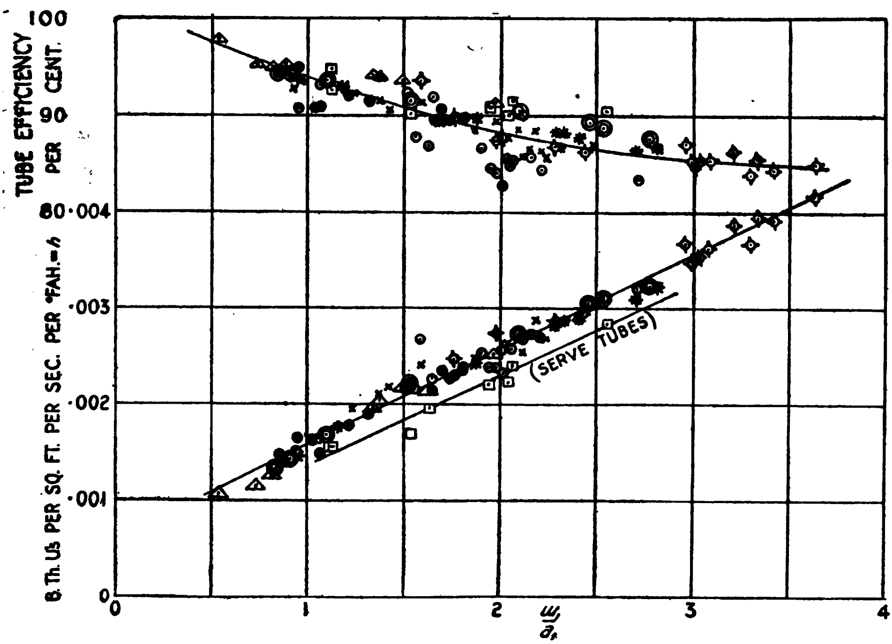


FIG. 6.—Rate of heat transmission and tube efficiency of locomotive boilers under test.

operation. In some cases a firebrick arch is placed in the furnace of a locomotive boiler to facilitate combustion. This should not materially affect the calculated temperatures, for although the arch would absorb some heat from the bed of fuel and from the flames it would radiate an equivalent amount to the fire-box.

It would also be noticed that the temperature of the furnace increases with $\frac{w_1}{a_1}$, as does also the smoke-box temperature,

but it is certain that the smoke-box temperature would not have risen to the extent shown had the fire-box temperature not increased. In the same figures were also plotted the weight of gas formed per pound of coal fired, and, generally speaking, this decreased as $\frac{w_1}{a_1}$ increased, as might be expected, though in some cases the values were somewhat erratic.

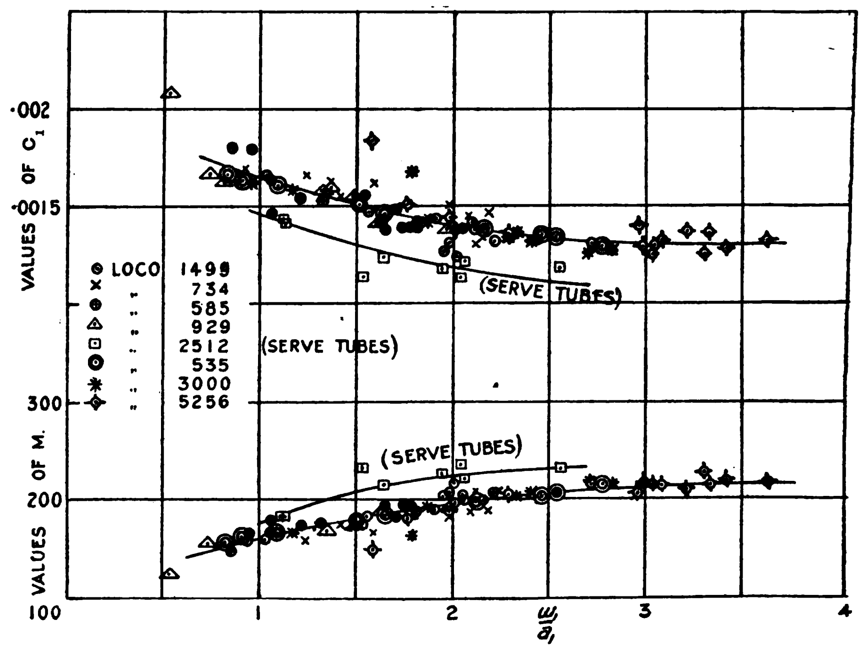


FIG. 7.—Values of "M" and "c_1" from locomotive boiler tests.

The rate of heat transmission through the tubes from the gases to the water was calculated for each test using the calculated fire-box temperatures T_1, and the values of w_1 obtained in the manner described. The mean difference of temperature between the gases and the water was taken to be

$$t_m = \frac{T_1 - T_2}{\log_e \left(\frac{T_1 - t_s}{T_2 - t_s} \right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where T_2 is the smoke-box temperature, and t_s the measured

steam temperature, taken to represent also the water temperature. Then the rate of heat transmission h per square foot of tube surface per second per degree Fahr. difference of temperature is

$$*h=\frac{w_1\times\cdot25\times log_e\left(\frac{T_1-t_s}{T_2-t_s}\right)\text{ B.Th.U.}}{A} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

A being the area of the tube surface (gas side) between the tube plates in square feet.

The various values of h calculated in this manner are shown plotted in Fig. 6 for all the tests, on the base $\frac{w_1}{a_1}$ and a mean line is drawn among the points. It will be noted that the straight line fairly represents the relation between h and $\frac{w_1}{a_1}$ within the limits of the experiments, except that the values for the Serve tubes† were slightly below those obtained with plain tubes, as might have been anticipated from the certainty that the ribs of the Serve tubes would be at a higher temperature than the other portions of the tube.

The calculated tube efficiencies $\frac{T_1-T_2}{T_1-t_s}$ are also plotted in Fig. 6, showing in general the same nature as those obtained from the model boilers, and illustrated in Fig. 69 *of Heat Transmission by Radiation, Conduction, and Convection.*

Again using the expression,

$$\frac{l}{m_1}=M\,log_e\frac{T_1-t_s}{T_2-t_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\text{where } M=\frac{s_1(c_1+C_2\,R\,\beta)}{c_1\,C_2\,R\,\beta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

the various values of M are shown in Fig. 7. Also, using $C_2=\cdot03$, the corresponding values of c_1 are also given in the same figure.

The method of calculation given on pp. 24 and 25 enables

* Here again it is not thought worth while to take into account the variable specific heat of the gases. The mean value is actually somewhat higher than ·25.

† Calculated on the surface in contact with the gas.

an estimate to be made of the various losses in the locomotive boilers. The author has calculated these as a percentage of the total heat of the fuel, and has illustrated the results by the diagrams in Figs. 8 and 9, referring to the locomotives Nos. 1499 and 734 respectively.

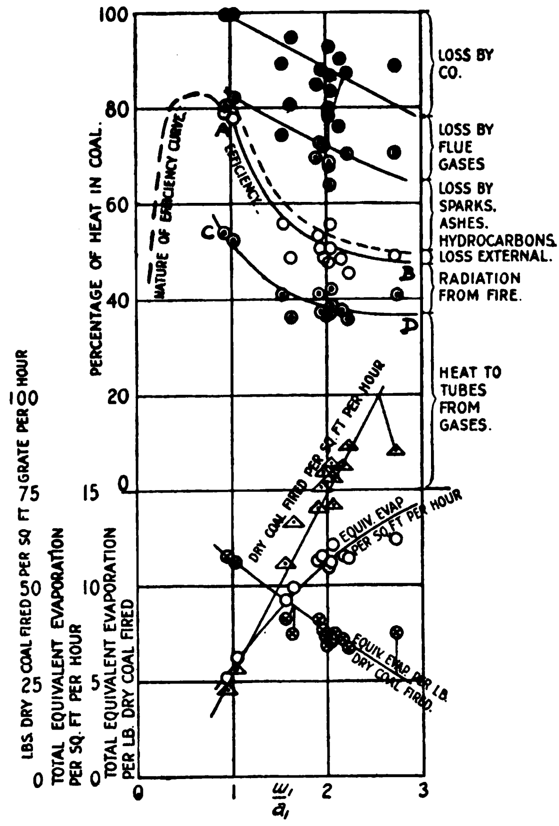


FIG. 8.—Data from tests on locomotive boiler 1499.

The lines A B in both figures indicate the mean boiler efficiencies* on the base $\frac{w_1}{a_1}$, the dotted line to the left representing the probable character of the efficiency curve as $\frac{w_1}{a_1}$

* Based on the gross calorific value of the fuel.

decreases below the experimental values. It will be noticed that the percentage loss of heat due to the formation of CO is rather large in locomotive No. 1499, Fig. 8, but it was found to be comparatively small for all the others ; in fact, similar

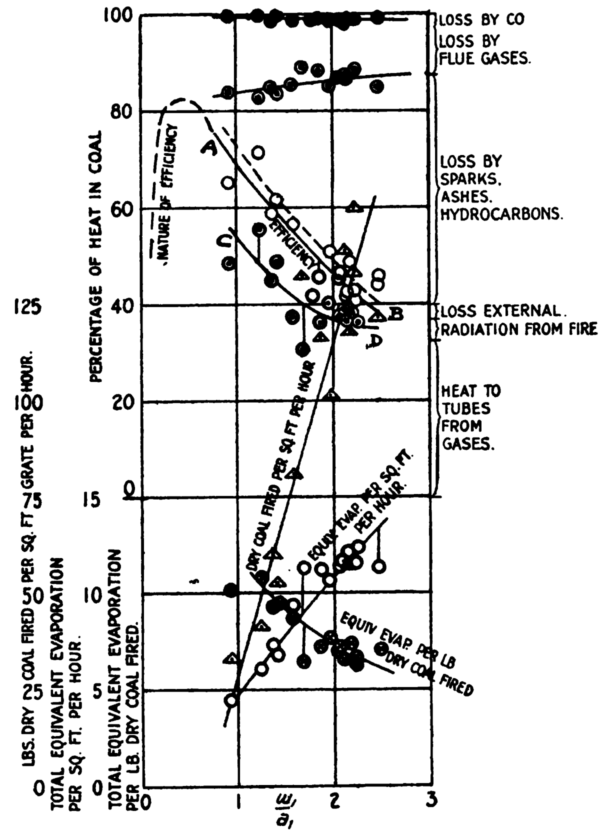


FIG. 9.—Data from tests on locomotive boiler 734.

to the values shown in Fig. 9. The outstanding feature in all the locomotives, however, was the large heat losses arising from sparks, cinders, unburnt hydrocarbons, etc., at high rates of working. Except for that, the boiler efficiency ought to fall but very little as the rate of working increases. It must also be noted that this loss practically vanishes in the neighbour-

hood of what would probably prove to be the maximum boiler efficiency.

For the purposes of comparison, there are also shown, in Figs. 8 and 9, the weight of dry coal fired per hour per square foot of grate, the total equivalent evaporation per pound of dry coal fired, and the equivalent evaporation per square foot of the total boiler heating surface, expressed in pounds of evaporation from and at 212° F.

Using the calculated furnace temperatures and the various values of the smoke-box temperatures, etc., the total heat transmitted to the water could be separated approximately into radiation from the fuel on the grate and heat-flow through

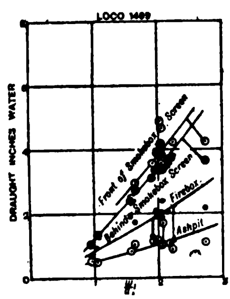


FIG. 10.—Draught in boiler of locomotive 1499.

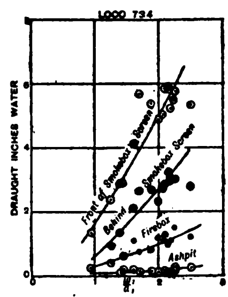


FIG. 11.—Draught in boiler of locomotive 734.

the tubes. The various plotted points and the mean lines C D, in Figs. 8 and 9, illustrate the results obtained, and it is evident that the calculated amount of heat radiated from the fuel on the grate was much less than the heat transmitted through the boiler tubes, particularly so at the higher rates of working.

The measured draughts in the ashpan, fire-box, and smoke-box, both in front of and behind the screen were obtained by means of U water gauges tested for accuracy of graduation and uniformity of bore of tubes, and the results were tabulated in the report mentioned on p. 20.

Figs. 10 and 11 show the draughts in the locomotives Nos. 1499 and 734 respectively in the ashpit, fire-box, and smoke-box, plotted on the base $\frac{w_1}{a_1}$. The measured draughts from the separate tests were too erratic to deduce values of f

directly, but such values were obtained from the various mean lines representing the draughts, as in Figs. 10 and 11. These various values of f are shown plotted in Fig. 27, p. 51.

Another series of locomotive experiments might be considered. These relate to tests* made on a consolidation type locomotive 958 of the Illinois Central Railroad, placed on a dynamometer. The principal boiler dimensions were as follow :—

Length of firebox, inside	108·25 in.
Width of firebox, inside	66·1 in.
Area of grate	49·5 sq. ft.
Heating surface of firebox	168 sq. ft.
Number of tubes	413
Diameter of tubes, outside	2 in.
,, ,, ,, inside	1·734 in.
Length between tube plates	195·2 in.
Heating surface of tubes (fire-side) . .	3094 sq. ft.
Total heating surface (including front tube plate)	3283 ,, ,,
$\dfrac{\text{Heating surface}}{\text{Grate area}}$	66·4
Area for gas-flow through tubes. . . .	6·77 sq. ft.
$\dfrac{l}{m_1}$	450

Full data is given in the Bulletin for each test, and the results have been dealt with by the writer according to the methods of calculation given on p. 24. The following Table 3 illustrates the character of the values obtained for the boiler efficiency and for the various losses of heat.

TABLE 3

Coal fired per hour per sq. ft. of grate. lbs.	Boiler efficiency, per cent. (based on gross calorific value of coal)	Loss by flue gases, per cent.	Calculated loss by cinders, sparks, hydrocarbons, etc., per cent.	Measured furnace temperatures (by optical pyrometer) °F.	Calculated furnace temperatures °F.	Smoke-box temperatures °F.	Steam temperature °F.
39·9	66·9	24·8	4·3	1407	1400	507	387
95	56·7	20 [illegible]	19·3	1597	1760	595	387
158	46·4	16·1	32·8	1785	2000	675	386
224	38·8	13·0	43	—	2180	703	386

* Bulletin No. 82, Engineering Experiment Station, Univ. of Illinois, U.S.A.

The calculated values of the rate of heat transmission h are shown plotted in Fig. 12 on the base $\frac{w_1}{a_1}$, and on comparing with the similar values in Fig. 6, p. 27, it would be seen that, on the whole, the values of h from Fig. 12 are slightly

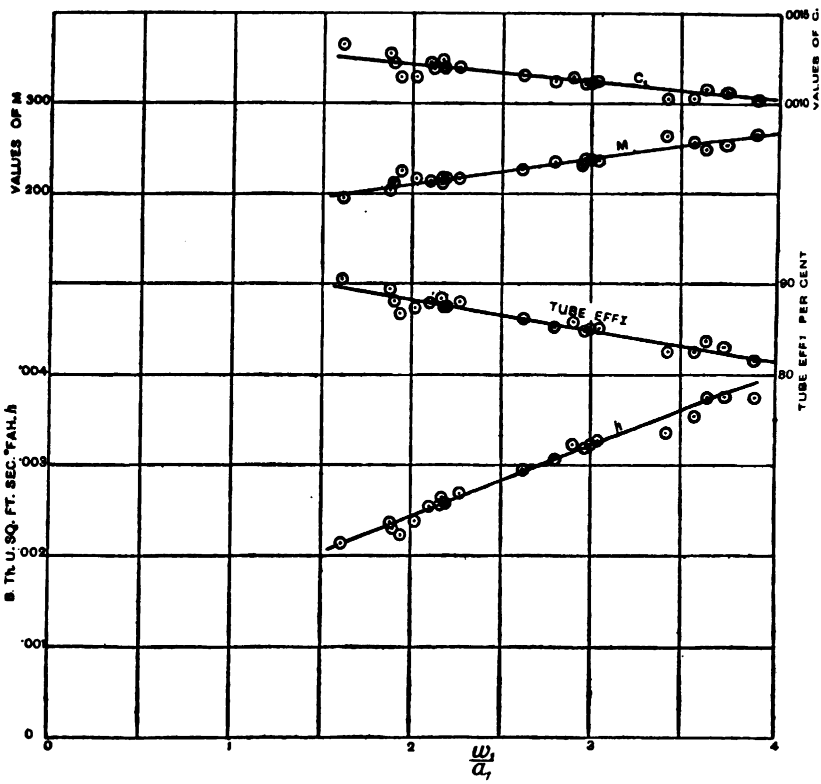


FIG. 12.—Rate of heat transmission and tube efficiency in Univ. of Illinois locomotive experiments.

below those from Fig. 6. The estimated tube efficiency is also plotted in Fig. 12 along with the calculated values of M and c_1 derived from equations 5 and 6, p. 29, again using ·03 as the value for C_2.

Tests on a Normand Boiler.—A series of tests made on a Normand water-tube boiler by Messrs. W. T. Ray and H. Kreisinger is described in Bulletin 403 of the U.S.A. Geological

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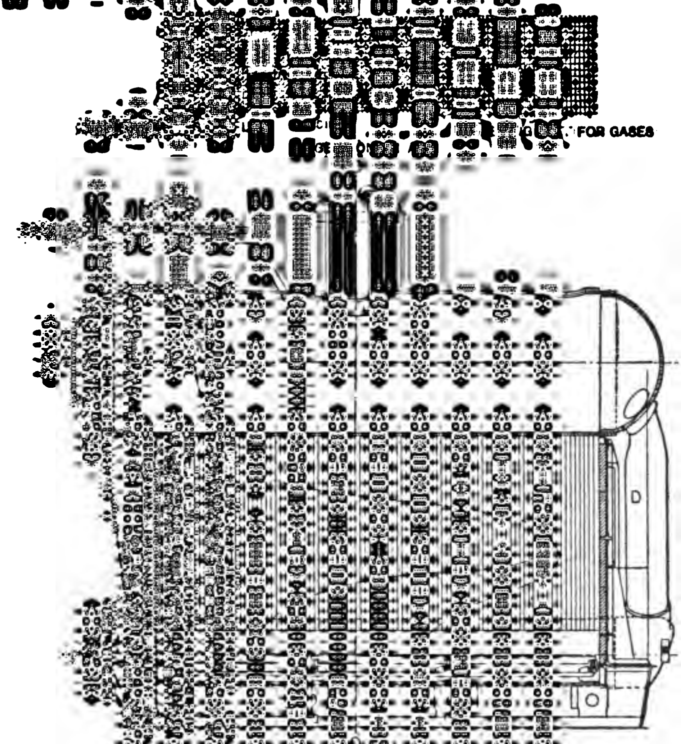

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quantity of sparks thus accounted for in no way corresponded with the amount of heat unaccounted for in the heat balance. The following dimensions are given in the Bulletin :—

Diameter of steam drum, in.	35·4
Length of steam drum, in.	150
Diameter of bottom drums, in.	10·75
Length of bottom drums, in.	141·75
Diameter of downcomers, in.	10·75
Number of tubes	1552
Outside diameter of tubes, in.	$1\frac{3}{16}$
Approximate length of tubes, in.	75·5
Total heating surface, sq. ft.	2776
Length of furnace, ft.	9·16
Width of furnace, ft.	6·4
Height of furnace from grate to steam drum, ft. .	4·65
Approximate combustion space above grate, cu. ft..	136
Distance from front of furnace to opening among tubes, ft.	5·5
Length of bars, in.	37·5
Average width of grate bars on top, in. . . .	·375
Average width of air spaces in grate, in. . . .	·56
Air spaces in grate (approximate), per cent . .	55
Area of grate, sq. ft.	58·6
Ratio of combustion space to grate area . . .	2·33

The whole boiler and furnace were enclosed in a cast-iron casing, which was lined with fire-brick at the front and rear of the combustion chamber, and with sheet asbestos along the nests of tubes. The gas temperature in the uptake was measured by a platinum, platinum-rhodium thermo-couple. The furnace temperature was measured with a Wanner optical pyrometer,* and for this purpose a hole $1\frac{1}{4}$-in. diameter was drilled in the furnace casing at the back of the boiler and about 3 ft. above the grate, the pyrometer being sighted on the furnace through this hole.

The boiler worked on the closed stokehold system, the air being supplied by a fan. Gas pressures were measured on all tests at the base of the stack, over the fuel bed, and in the ashpit, and were recorded in the report.

In some of the tests coal was used, and in others large and

* Refer to p. 84 of *The Measurement of Steady and Fluctuating Temperatures* for a description of this type of pyrometer.

small briquets were burned, made from the same class of coal as was used in the other tests mentioned. The fuel was analysed for each test, and varied only slightly in composition

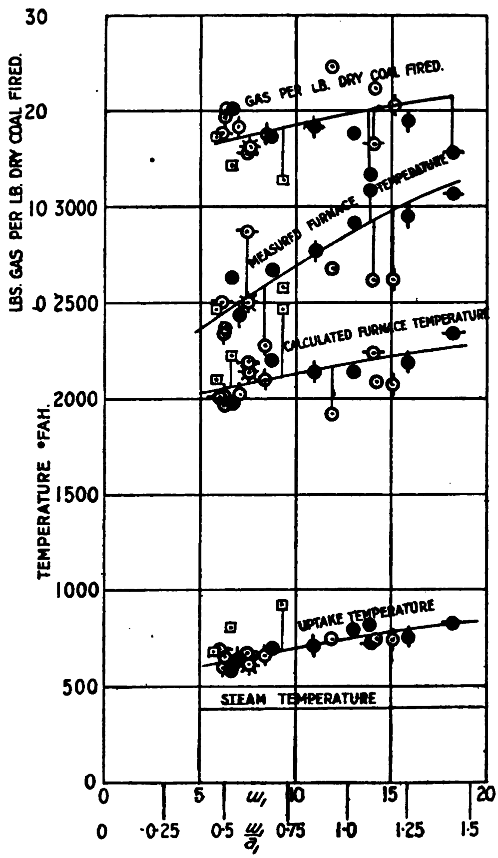


Fig. 14.—Temperatures in Normand water-tube boiler under test.

between the various tests. For the coal in the first test, the following composition was given :—

Composition of Dry Coal.

Carbon . .	82·58%	Nitrogen .	1·45%
Hydrogen .	4·55%	Ash . .	5·55%
Oxygen . .	5·10%		

Calorific value of dry coal (gross) 14,845 B.Th.U. per pound. From the tabulated results the author has calculated the weight of the gases per second w_1 by the methods illustrated in the calculations on p. 24. Furnace temperatures were also

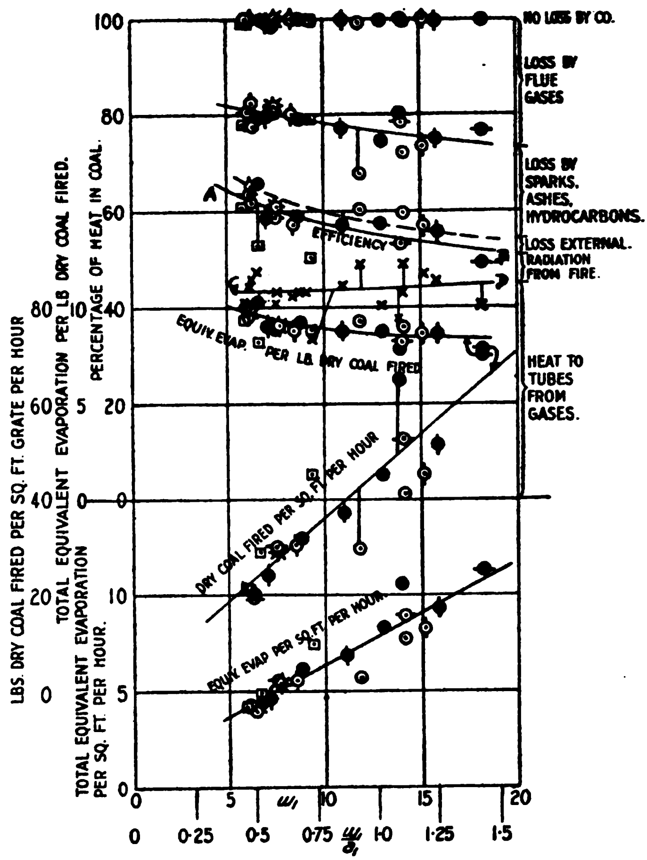


FIG. 15.—Data from Normand water-tube boiler under test.

calculated by the method shown on p. 25. These values are plotted in Fig. 14, as well as the measured furnace and uptake temperatures, and it is seen that the measured furnace temperatures were probably from about 400° F. to about 800° F. too high. In succeeding calculations the calculated furnace temperatures have been used. Here, again, there is an evident

rise of the furnace temperature as the rate of working increased, which caused, it is certain, a part of the rise shown in the uptake temperature. The weight of gases formed per pound of dry coal fired is also shown in Fig. 14, and is found to be somewhat erratic.

Calculations have also been made of the various losses, and these, along with boiler efficiencies,* have been plotted in

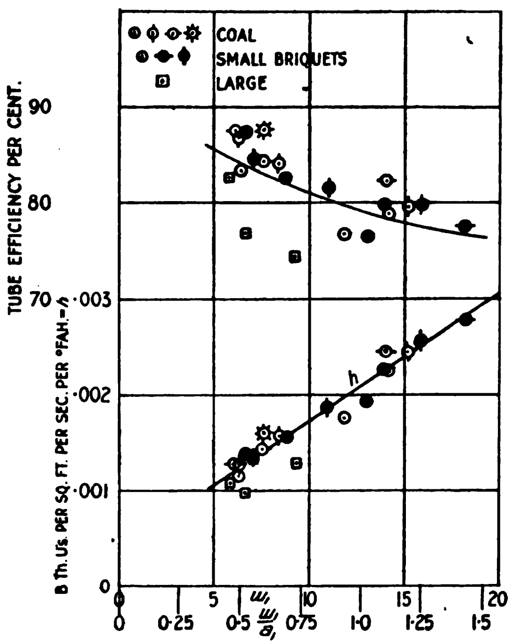


FIG. 16.—Rate of heat transmission and tube efficiency of Normand water-tube boiler under test.

Fig. 15. Here, again, it is seen that the losses due to sparks, hydrocarbons, ashes, etc., must have been large, although the amount of heat lost externally to the atmosphere by radiation, etc., may have been larger than is indicated in Fig. 15. The loss by the formation of CO is practically negligible. The mean line C D indicates the relation between the calculated amount of heat given to the boiler by radiation from

* Based on the gross calorific value of the fuel.

the fuel on the grate, and the amount given by the gases in passing among the tubes.

In the same figure is also shown the weight of dry coal fired per hour per sq. ft. of grate, as well as the total equivalent evaporation from and at 212° F. per pound of dry coal fired, and the equivalent evaporation per sq. ft. of total heating

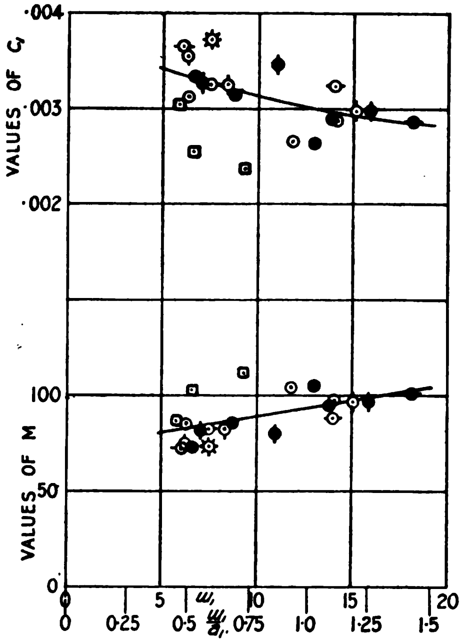


FIG. 17.—Values of "M" and "c_1," from Normand water-tube boiler under test.

surface per hour. On the base line is shown the approximate values of $\frac{w_1}{a_1}$ where a_1* was roughly calculated as the average area in sq. ft. available for the flow of the gases among the tubes.

In Fig. 16 the calculated values of the rate of heat transmission h are shown plotted, calculated from the expression 4 on p. 29. These values of h may be compared with the

* Refer to p. 76 for an example of the method used for finding the average value of a_1.

corresponding results from the air heater shown in Fig. 75, p. 173 of *Heat Transmission by Radiation, Conduction, and Convection*; and it will be seen that the latter come out somewhat higher than in Fig. 16; but in both cases the values obtained

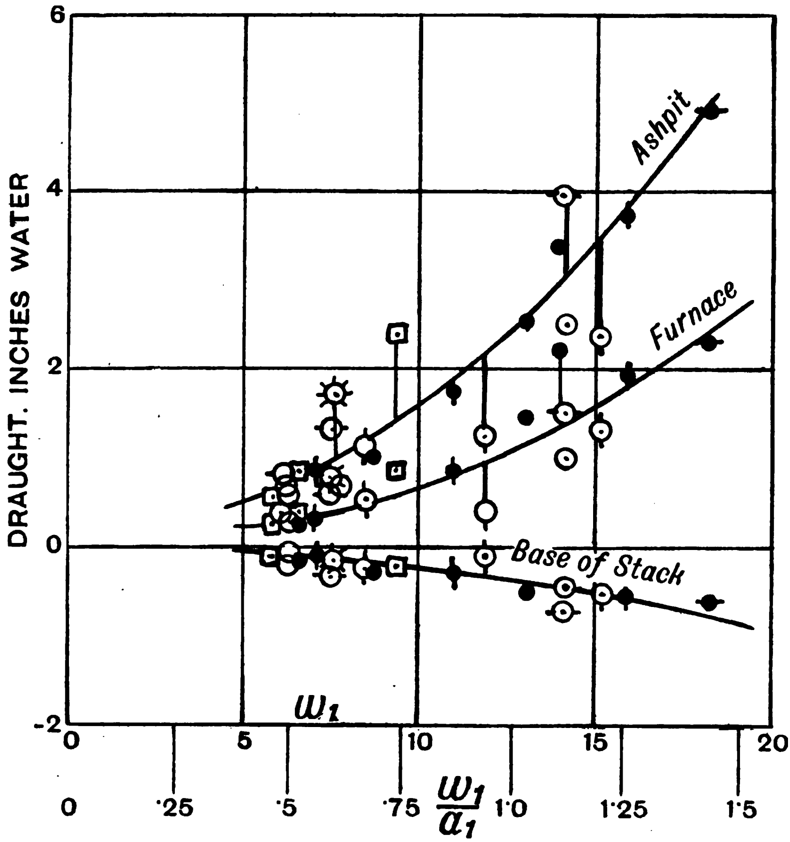


FIG. 18.—Gas pressures in Normand water-tube boiler.

signify that the rate of heat transmission with the gas-flow across tubes set zigzag is much greater than when the flow occurs through plain tubes. The tube efficiencies $\frac{T_1 - T_2}{T_1 - t_s}$ are also shown in the same figure.

Assuming the average length of the path of the gases among

the tubes l to be equal to the length of the furnace, values of M in the equation

$$\frac{l}{m_1}=\mathrm{M}\,log_e\frac{\mathrm{T}_1-t_s}{\mathrm{T}_2-t_s}$$

are plotted in Fig. 17. Again, using the value for

$$\mathrm{C}_2=\cdot 03 \text{ in } \mathrm{M}=\frac{s_1\,(c_1+\mathrm{C}_2\,\mathrm{R}\,\beta)}{c_1\,\mathrm{C}_2\,\mathrm{R}\,\beta},$$

the calculated values of c_1 are also given in Fig. 17.

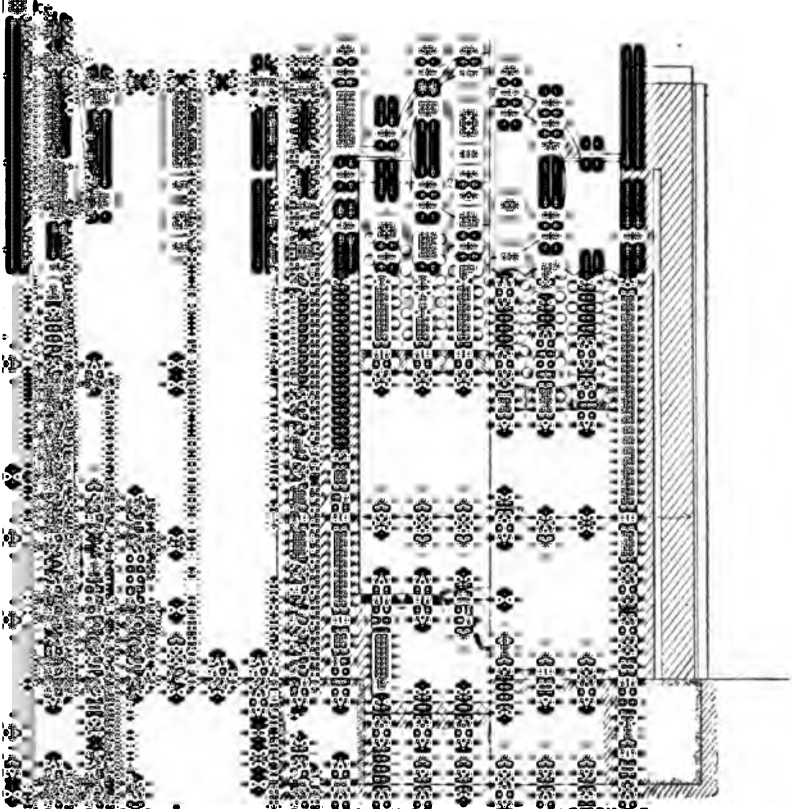
The measured pressures in the ashpit, furnace and at the base of the stack are shown plotted in Fig. 18.

Tests on a Heine Boiler.—A very large number of complete boiler tests made on different water-tube boilers by Messrs. L. P. Breckenridge, H. Kreisinger, and W. T. Ray, are recorded in Bulletin 23,* issued by the Bureau of Mines, Washington. A few of these tests have been selected by the author where one class of coal was burned at different rates under a Heine water-tube boiler. This boiler is shown in Fig. 19, and in the tests to be considered the coal was fired by an underfeed stoker. The principal dimensions of the boiler were as follows :—

Length of steam drum, ft.	$21\frac{7}{12}$
Inside diameter of drum, in.	42
Number of tubes	116
Internal diameter of tubes, in.	3·26
External diameter of tubes, in.	3·50
Water-heating surface of tubes, sq. ft.	1897
Total water-heating surface, sq. ft.	2031
Length of furnace from front wall to bridge wall, ft.	6·3
Width of furnace between side walls, ft.	6·2
Cross-sectional area between tubes a_1, sq. ft.	11·2

The bottom row of tubes were enclosed in C-shaped tiles for their entire length, except for 30 inches in the rear of the boiler, where an opening was left to admit the gases round the tubes. A similar opening was left in the baffle on the top row of tubes at the front of the boiler to allow the gases to leave the tubes, which then flowed under the drum to the uptake at the back of the boiler ; this opening was 18 inches long. Induced

* "Steaming Tests on Coals, and Related Investigations," September, 1904, to December, 1908.



page 42.

draught was used, produced by a fan placed between the uptake and the chimney stack.

The duration of each test was about nine hours, and the coal and water were measured as accurately as possible, as well as the dryness fraction of the steam. The calorific value and composition of the coal in each test was determined, small differences only arising between the different tests here dealt with. The gross calorific value of the dry coal would average about 15,000 B.Th.U. per pound, and the average composition of the coal was as follows :—

Carbon . .	84·4%	Nitrogen . .	1·43%
Hydrogen .	4·73%	Sulphur . .	0·95%
Oxygen . .	3·30%	Ash . . .	5·21%

Elaborate provision was made for collecting an average sample of the flue gases in the boiler uptake.

The uptake temperature of the gases was measured with the thermometer bulb inside the arrangement for collecting an average sample of the gases, and thus it was practically screened from external radiation. In the calculations which follow it has been taken that the temperature of the gases leaving the tubes would be about 30° F. higher than in the uptake. The furnace temperature was measured with a Wanner optical pyrometer.

The author has estimated the weight of gases flowing per second w_1 by the methods explained on p. 24, and also calculated the furnace temperatures according to the principles explained on p. 25. In this case, however, these estimates of furnace temperature were likely to be too low, because they were based on the assumption that the radiation from the fire was completely absorbed by the cold tube surfaces, whereas, in this case the bottom row of tubes was enclosed in firebrick, the surface of which would have a somewhat higher temperature than that of the tubes, and would also reflect some of the heat radiations emanating from the fuel on the grate.

These various temperatures are shown plotted on the base $\frac{w_1}{a_1}$ in Fig. 20. It would be seen that the measured furnace temperatures were probably not far from the correct values

in these experiments, but in the following calculations the calculated furnace temperature has been used. The weight of gases formed per pound of dry coal fired is also shown in the same figure, and it would be noted that these values are very erratic.

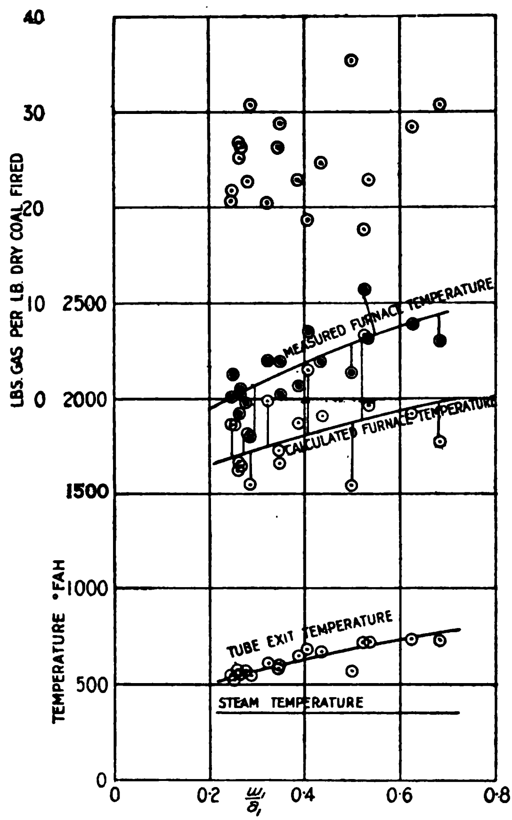


FIG. 20.—Temperatures in Heine water-tube boiler under test.

The boiler efficiencies* are shown plotted in Fig. 21, and the mean line A B drawn in, along with the various calculated heat losses. It is evident that, whilst the heat lost by the flue gases leaving the boiler was large, that lost by sparks, hydrocarbons, etc., could not have been very appreciable. On the

* Based on the gross calorific value of the fuel.

same base in Fig. 21 are also shown the weights of dry coal fired per hour, the total equivalent evaporation from and at 212° F. per pound of dry coal fired, and the equivalent evaporation per square foot of total boiler heating surface per hour, as well as the measured draught in the furnace and the uptake.

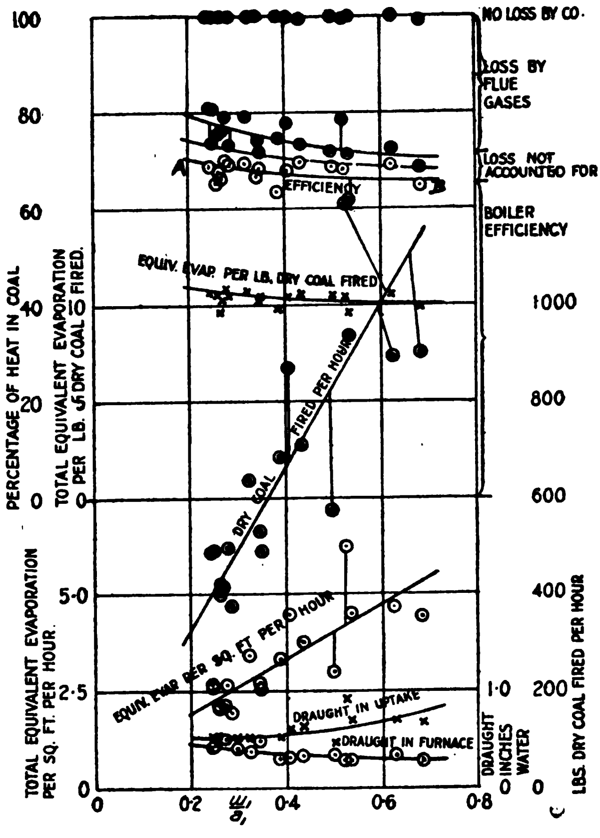


Fig. 21.—Data from Heine water-tube boiler under test.

Using the calculated furnace temperatures, the rate of heat transmission h through the tubes between the gases and the water are plotted in Fig. 22, the calculations being made according to the equation 4, p. 29. The calculated tube efficiencies are also shown in this figure, and it would be seen

that these efficiencies are somewhat low in value, particularly at the higher rates of working.

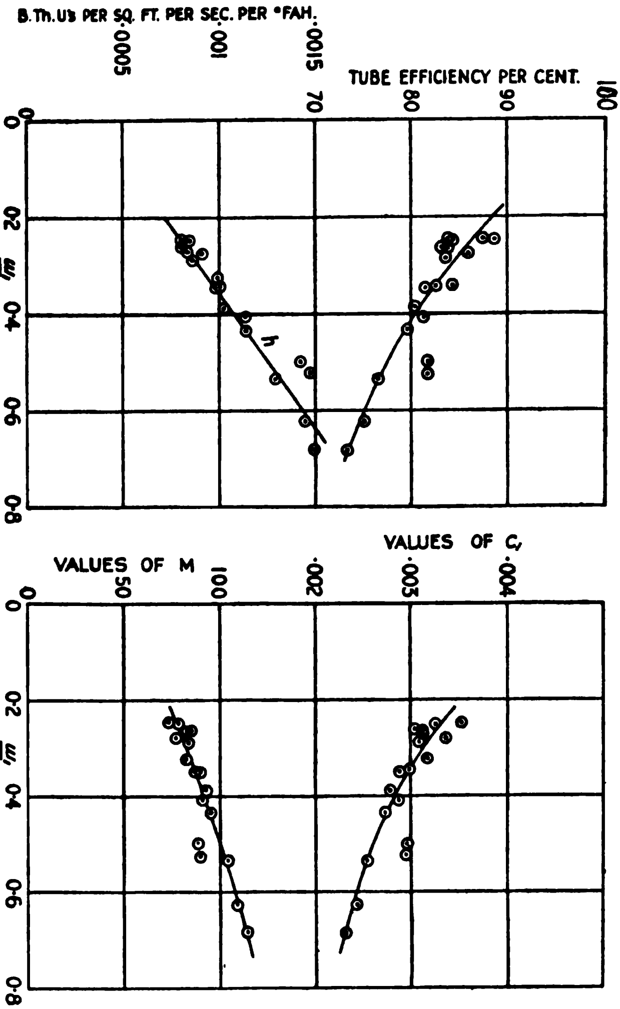


FIG. 22.—Rate of heat transmission, tube efficiency, "M" and "c_1," from Heine water tube boiler under test.

The value of the hydraulic mean depth m_1 was also calculated, and M obtained for each test by using the equation 5, p. 29. These values are plotted in Fig. 22, as also are the

corresponding values of c_1 calculated after inserting the value for $C_2 = \cdot 03$ in equation 6, p. 29.

The mean lines representing the rates of heat transmission for most of the boiler experiments considered are collected together in Fig. 23, and for comparison the corresponding curves of rates of heat transmission are reproduced for the

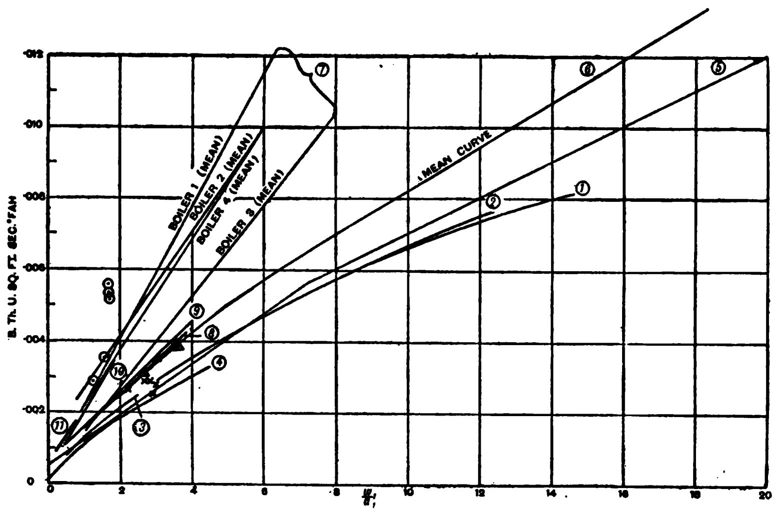


FIG. 23.—Rates of heat transmission from gas to water.

⊙ Nicolson-Longridge (Flue with firebrick plug). m_1 for heat-flow 1·9 in. Mean gas temp. 1500 to 2100° F.
× Nicolson-Longridge (Evaporator). m_1 for heat-flow 0·709 in. Mean gas temp. 860 to 1060° F.
△ Nicolson-Longridge (Economiser). m_1 for heat-flow 0·479 in. Mean gas temp. 430 to 540° F.

1. Royds. m_1 for heat-flow 0·248 in. Mean gas temp. 330 to 390° F.
2. Royds and Campbell. m_1 for heat-flow 0·248 in. Mean gas temp. 300 to 430° F.
3. Royds. m_1 for heat-flow 0·54 in. Mean gas temp 550° F.
4. Josse. m_1 for heat-flow 0·226 in. Mean gas temp. 100 to 140° F.
5. Nusselt. m_1 for heat-flow 0·217 in. Mean gas temp. 90° F.
6. Jordan. m_1 for heat-flow 0·1265 in. to 0·492 in. Mean gas temp. 200 to 590° F.
7. Small boilers. m_1 for heat-flow 0·044 in. to 0·0732 in. Mean gas temp. 320 to 1000° F.
8. Bell. m_1 for heat-flow 0·5 in. Mean gas temp. 1200° F.
9. Loco-boilers. m_1 for heat-flow 0·325 in. to 0·5 in. Mean gas temp. 900 to 1400° F.
10. Normand water-tube boiler. m_1 for heat-flow 0·72 in. Mean gas temp. 1150 to 1500° F.
11. Heine. m_1 for heat-flow 1·4 in. Mean gas temp. 900 to 1300° F.

small plant considered in Chapter IV of *Heat Transmission by Radiation, Conduction, and Convection*. The plotted points shown in this figure were derived by the author from the tests made by Mr. Longridge on Dr. Nicolson's experimental boiler. The circular points refer to the plug flue, and show relatively high values, the probable reason being that the gases lost a certain

amount of heat to the firebrick plug which radiated the heat to the colder surrounding surfaces. The points shown as crosses refer to the evaporator, and the triangles to the results obtained from the economiser. In these two cases it would be seen that the points agree fairly well with most of the other experiments considered.

It may be stated, therefore, that notwithstanding the approximate nature of the boiler calculations, the results

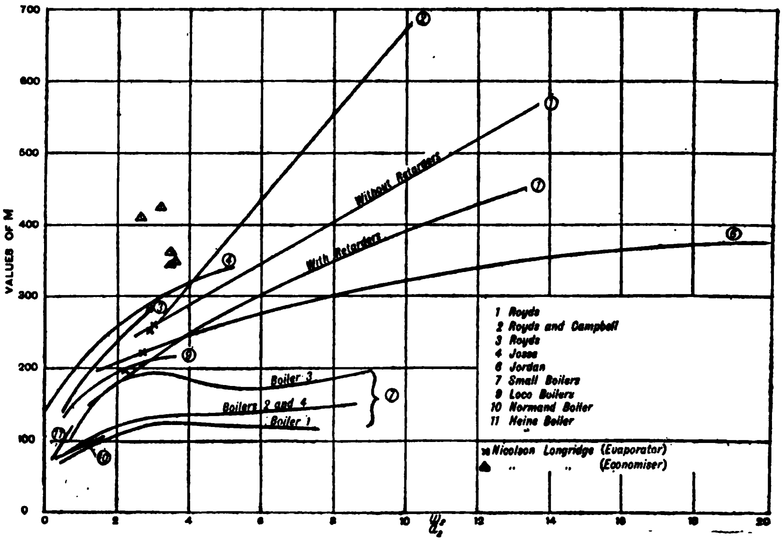


FIG. 24.—Values of M from equation 5, p. 29.

obtained from the several boilers agree fairly well with the experiments made on the various small apparatus, and that, whilst the rate of heat transmission h between the gases and water does not appear to be influenced greatly by the temperature of the gases, it would seem that the diameter of the tubes, or the hydraulic mean depth of the gas passage, has an appreciable influence.

From the various mean lines in Fig. 23 there is every indication that the rate of heat transmission h tends towards a value not far from zero as the value of $\frac{w_1}{a_1}$ approaches zero. There is also reason to believe that the gradual curvature of

some of these lines is due to the smoothness of the tubes used. With rough tubes it is likely that the line of heat transmission would remain nearly straight until near to the zero value of $\frac{w_1}{a_1}$, when it would probably drop very rapidly to nearly zero.

Another factor affecting the curvature of the lines in Fig. 23 is the slight rise of the tube temperature as the rate of heat transmission increases.

For purposes of comparison the various mean lines representing the calculated values of M in equation 5, p. 29,

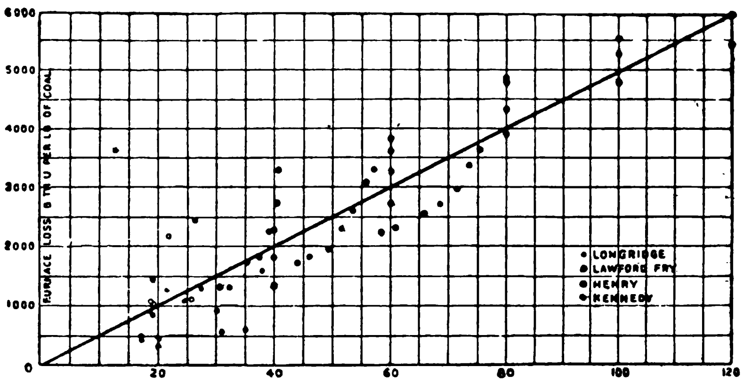


FIG. 25.—Furnace loss in relation to rate of firing.

are reproduced in Fig. 24. Whilst these lines appear to give widely different values of M, a fairly large variation in the value of M does not usually affect the calculated fluid temperatures to any large extent.

Finally, it may be stated that the results of the boiler experiments plainly indicate that one of the most important problems affecting the efficiency of boilers at high rates of working is the efficient combustion of the fuel.

The late Professor J. T. Nicolson* proposed that the furnace losses due to incomplete combustion, loss of fuel, etc., could be represented approximately by,

$$\text{B.Th.U. lost per lb. coal fired} = 50\,F.$$

* *Trans. Inst. Engs. and Shipbds. in Scotland,* Vol. LIV, 1910-11.

where F is the rate of combustion in lbs. of coal per sq. ft. of grate per hour. This equation was based upon the results shown in Fig. 25. It is seen that some of the points in this figure lie very wide of the mean line, and the equation can only be taken as a rough general approximation. Nor is it likely that these results in Fig. 25 were derived by the author's method on p. 24.

From the results shown plotted in Fig. 26 Professor Nicolson

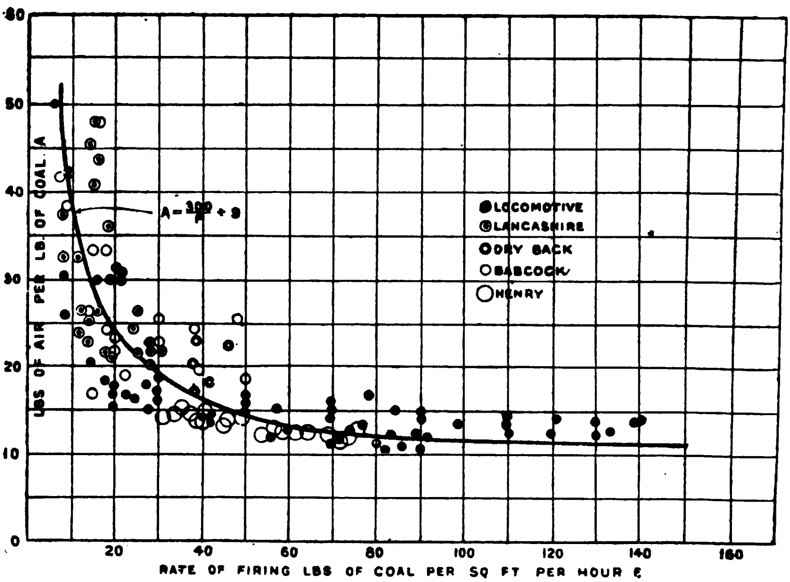


FIG. 26.—Air supply per pound of coal and rate of firing.

also proposed that the air supply A per lb. of coal fired in ordinary furnaces could be expressed approximately by,

$$A=\frac{300}{F}+9 \text{ lb. per lb. coal fired.}$$

Here again it is obvious from the graph that the actual value of A might be widely different from the value given by the above formula, and in any case it is hardly likely that the plotted values are deduced according to the methods used by the author on p. 24.

Theory suggests that the rate of combustion in a firebrick

furnace is only limited by the rate at which air can be supplied to the furnace and intimately mixed with the burning fuel before the gases come into contact with cold surfaces. This was confirmed by experiments made by Professor A. L. Mellanby on an oil-fired firebrick combustion chamber at

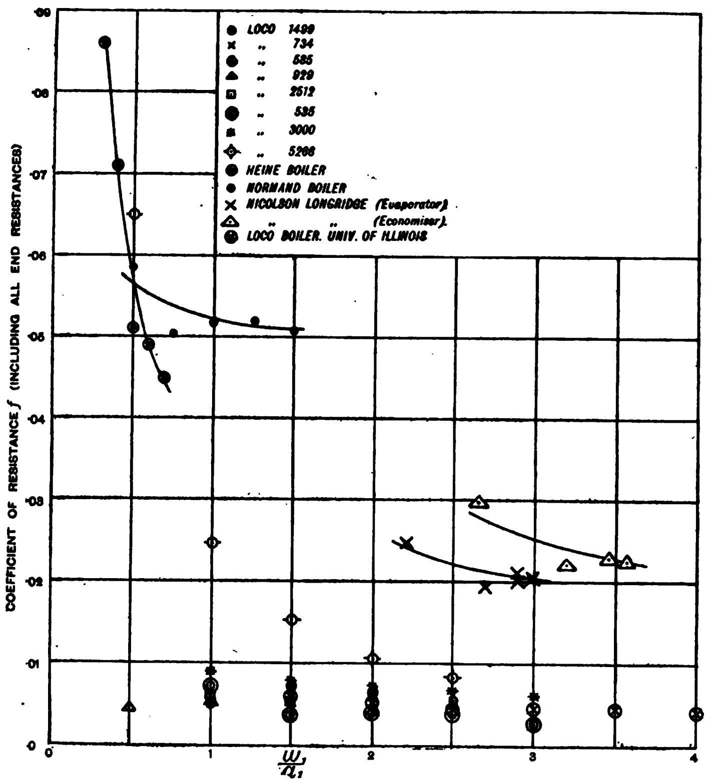


FIG. 27.—Coefficient of resistance f_1 and rate of gas flow.

the works of Messrs. Scotts' Shipbuilding and Engineering Co. Ltd., Greenock. The inside dimensions of the chamber were 18 in. × 18 in. × 4 ft. 6 in. long. A firebrick bridge was built about one foot high at about two-thirds along the length, and behind this was arranged a refractory tube closed at the inner end into which a Féry radiation pyrometer was sighted to measure furnace temperatures. The oil burners used were of the Wallsend-Howden type supplied with heated heavy

oil under pressure, the specific gravity of the oil at 69° F. being 0·906. A water-cooled sampling pipe was arranged to project into the combustion chamber for the withdrawal of samples of the gases from the front, middle, and back during each test.

The most important data from these experiments are given in the following table. The second and third tests shown in the table were made with a firebrick grid in the combustion chamber and it is seen that about 22 lbs. of oil per hour were burned efficiently per cubic foot of chamber volume, with only a moderate amount of excess air supplied. This is a much higher rate of combustion than is sometimes assumed to be possible in such a furnace.

Oil pressure, lbs. sq. in.	—			140			100		
Oil temperature, °F.	152			125			119		
Furnace temperature, °F.	2910			3000			3180		
Oil per hour, lbs.	144·5			145·4			198·5		
Oil per hr. per cu. ft., lbs.	16·1			16·2			22		
	Front	Middle	Back	Front	Middle	Back	Front	Middle	Back
Composition of gases CO_2	11·57	10·02	12·65	7·6	6·5	13·3	5·46	5·58	12·0
Composition of gases O_2	3·68	3·23	2·83	1·6	1·76	1·54	8·91	7·83	4·6
Composition of gases CO	1·24	1·08	0·79	0	0	1·10	2·3	2·82	0
Remarks	—	—	—	Very dirty	Very dirty	Dirty	Dirty	Clear	Clear
				Firebrick grid in combustion chamber.					

Flow of Gas through Boiler Tubes and the Coefficient of Resistance.—By the method illustrated in Fig. 82, p. 181 of *Heat Transmission by Radiation, Conduction, and Convection*, the writer attempted to determine the values of the coefficient of resistance to the gas flow, f, for the various boilers discussed from pp. 2 to 47; but in each case the plotted points lay in such erratic positions as to prevent any really useful values of f being obtained and an alternative method was adopted. For each boiler the draughts or gas pressures were plotted on a base $\frac{w_1}{a_1}$ as in Figs. 10 and 11, and smooth curves drawn in. For definite values of $\frac{w_1}{a_1}$ the drop of pressure through the tubes were then obtained from the curves and tabulated. The corresponding inlet and outlet gas temperatures were also tabulated

from the curves in Figs. 4 and 5. At each particular value of $\frac{w_1}{a_1}$ use was made of the equations,

$$\frac{P \times \Delta P}{\tau} = f_1 \times \frac{53 \cdot 2}{64 \cdot 4} \times \frac{l}{m_1} \times \left(\frac{w_1}{a_1}\right)^2 \quad \ldots \ldots \ldots \ldots \quad (1)$$

and also,

$$\Delta P = \delta P + \frac{1}{64 \cdot 4 \rho_1}\left(\frac{w_1}{a_1}\right)^2$$

$$= f \times \frac{53 \cdot 2}{64 \cdot 4} \times \frac{l}{m_1} \times \frac{\tau}{P} \times \left(\frac{w_1}{a_1}\right)^2 + \frac{1}{64 \cdot 4 \rho_1}\left(\frac{w_1}{a_1}\right)^2 \quad \ldots \quad (2)$$

where the various symbols have the significance stated on p. 18. The values of $\frac{l}{m_1} = \frac{A_1}{a_1}$ were calculated and a small addition made, if necessary, to allow for the enclosing walls.

The various values of f_1 and f obtained are shown plotted in Figs. 27 and 28 respectively on the base $\frac{w_1}{a_1}$. Except for loco. 5266, for which boiler the measured draughts were extremely erratic, the locomotive boilers gave quite low values of f, and are similar to the values of f derived from the author's experiments.

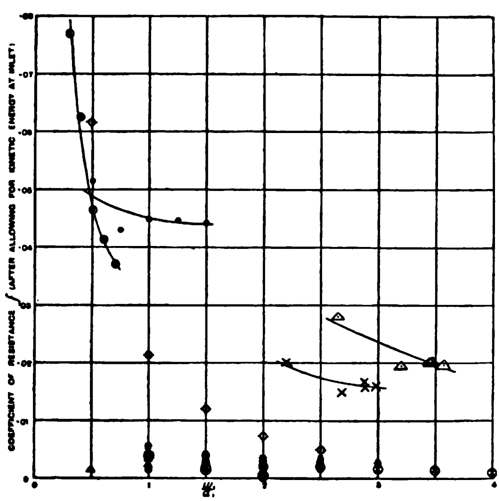


FIG. 28.—Coefficient of resistance f and rate of gas flow

The derived values of f_1 and f for the Nicolson-Longridge tests, and also for the Normand boiler and the Heine boiler previously described, are also shown plotted in Figs. 27 and 28 respectively. In all three cases the gases were in contact with the outside of the tubes and the water was inside, and the values of f are seen to be much greater than for the author's experiments and for the locomotive boilers, where the gases were in contact with the inside of the tubes. The probable reason for these differences, however, is seen when the conditions are considered. In the Normand type of boiler, for instance, the gases flow almost directly across tubes set diagonally as illustrated in Fig. 29. Considering the two streams A and B it will be noted that these will acquire a maximum velocity at the narrowest cross-section, say A and B respectively, and that immediately after passing these sections a sudden enlargement occurs and the two streams impinge on each other, with the probable result that a large portion of the kinetic energy of the streams at A and B will be lost by shock, and intense eddying motions will be set up in the space between the tubes C, D, and E. A similar action then takes place as the gases flow between the tubes D and E and also at each row of tubes from inlet to outlet. To produce the increase of kinetic energy at each passage between the tubes requires a corresponding amount of pressure energy, and it is not surprising to find, therefore, that the apparent coefficient of resistance f is much greater than for the flow through a smooth tube. In the Nicolson boiler and in the Heine boiler, although the gases flow along the length of the tubes, it is obvious that they must also flow across the tubes to some extent at both the inlet and the outlet, and therefore much the same action as is described above would occur at these places. No doubt this accounts to some extent for the apparently large value of f for the Nicolson experimental boiler.

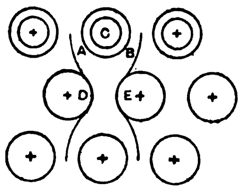


FIG. 29.—Flow of gases across tubes.

Relation between the Amount of Fuel Fired and the Rate of Evaporation in Boilers.—It is obvious from a consideration of Figs. 8 and 9 that, within certain limits, the weight of

the products of combustion, w_1 lb. per sec., increases with the amount of fuel fired into a boiler furnace. Also the amount

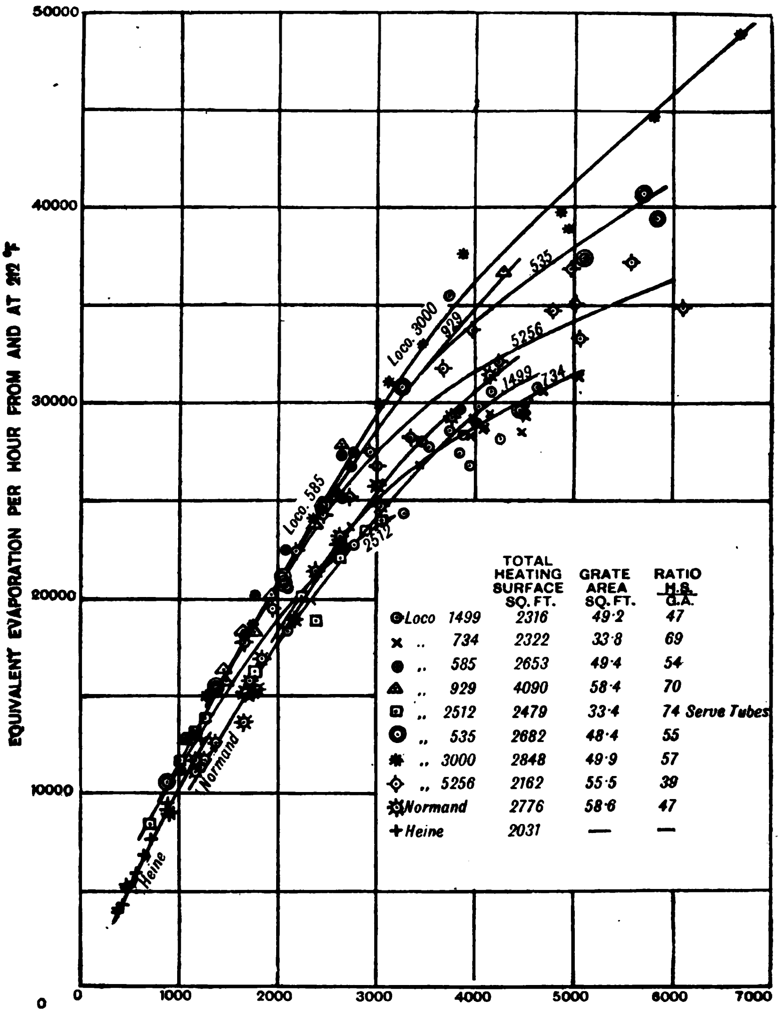


FIG. 30.—Relation between evaporation and coal fired in boilers.

of water evaporated increases with the rate of heat transmission. Thus it would be expected that the evaporation per hour when plotted against the weight of fuel fired per hour

would give curves similar in character to those given in Fig. 23. That this is so is shown by the results of the boiler tests previously considered plotted in Fig. 30. Here the evaporation in lbs. per hour from and at 212° F. is plotted against the weight of dry fuel fired per hour for the locomotive tests and for the tests on the Normand and the Heine water-tube boilers. It is also obvious that similar curves would be obtained for these boilers if the evaporation per hour per square foot of heating surface were plotted against the weight of fuel fired per hour per square foot of grate, except that these curves would lie more distinctly apart according to the relative values of the ratio of the heating surface to the grate area.

The Influence of Boiler Scale on the Efficiency.—There are few reliable boiler tests available demonstrating the influence of scale on the boiler efficiency. Some experimenters from a few isolated tests have found comparatively little loss of efficiency due to scale, whilst others again have apparently noted an appreciable loss. It is somewhat difficult, however, to make deductions from a few isolated experiments on boilers, because there are so many incidental sources of error in boiler experiments which may completely vitiate the results. An extensive series of experiments, however, have been made by E. C. Schmidt* on locomotive boilers extending over several years. Briefly stated his conclusions were that the loss of *heat transmission* due to scale $\frac{1}{8}$ in. thick may vary in individual cases from insignificant amounts to as much as 12%; the loss increasing somewhat with the thickness, but not directly as the thickness. The mechanical structure of the scale has a greater influence on the loss than the thickness, and chemical composition has no influence except in so far as it affects the structure of the scale.

It does not follow, however, that the efficiency of a boiler is affected to anything like the extent of 12%. A reference to example IV, p. 85, shows that the probable influence of a $\frac{3}{16}$-in. scale on the rate of heat transmission, h, for a locomotive type of boiler working at a high rate is to reduce it from the value, say, ·003 to ·00275, or a reduction of about

* *Railroad Gazette*, Vol. 33, p. 408. University of Illinois, Bulletin No. 11.

8·3%; but the efficiency of the boiler is only reduced by about 1%. Similarly, for a boiler working at a low rate the value of h was reduced from ·00115 to ·00111 or a reduction of 3·5%, whilst the efficiency of the boiler was estimated to fall only 0·7%. In any case, the influence of boiler scale is likely to be greater the higher the rate of working.

Fans and Blowers.—To operate boilers at high rates of evaporation it is generally necessary to adopt forced or induced draught, produced by means of fans, blowers, or steam jets. There are certain fundamental laws relating to fans and blowers of the centrifugal type which are sometimes useful

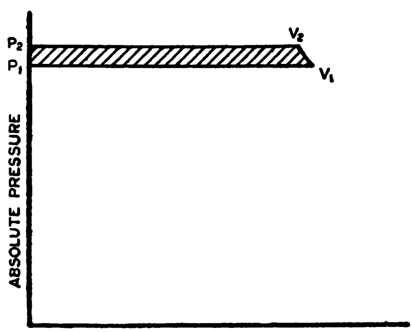


FIG. 31.—Area represents work done by fan or blower.

in estimating the performance of such plant. When operating under given conditions as regards the external resistances to the flow of the air or gases, the following relations hold approximately over a fairly large range for a given fan or blower.

1. The quantity of air discharged varies as the speed of rotation.
2. The pressure produced varies as the square of the speed.
3. The power required varies as the cube of the speed.
4. In a series of fans of the same type, having similar dimensions, same blade or vane angles, and rotating with the same blade-tip speeds under similar conditions of external resistance, the power required and the quantity of air discharged vary as the square of the linear dimensions.

Referring to Fig. 31 if the law of compression is $PV^n=$

constant, where P is the pressure, V the volume, and n a constant, then the work done, W, on the gases per lb. is given by

$$W=\frac{n}{n-1}c\tau_1\left\{\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}-1\right\}\quad . \quad . \quad . \quad (1)$$

where τ_1 is the absolute temperature at inlet.

$$c=\frac{PV}{\tau}\quad . \quad . \quad . \quad (2)$$

For one pound of gas $c=53{\cdot}2$ nearly with ft., lb., deg. Fah. units.

For a single stage fan or blower under ordinary conditions of operation it is evident from Fig. 31 that this work W may be approximately estimated by

$$W=(P_2-P_1)\frac{(V_2+V_1)}{2}\quad . \quad . \quad . \quad (3)$$

For example, with an induced draught fan compressing gases at 500° F. from 13·7 to 14·7 lb. per sq. in., and taking $n=1{\cdot}4$.

By equation 1, $W=\frac{1{\cdot}4}{{\cdot}4}\times 53{\cdot}2\times(500+460)\left\{\left(\frac{14{\cdot}7}{13{\cdot}7}\right)^{{\cdot}286}-1\right\}$

$=3545$ ft. lb. per lb. gases.

Now, $\frac{V_2+V_1}{2}=\frac{53{\cdot}2\times 960}{14{\cdot}2\times 144}=25$ cub. ft. per lb. nearly.

Thus, by equation 3

$$W=(14{\cdot}7-13{\cdot}7)\times 144\times 25$$
$$=3595 \text{ ft. lb. per lb. gases,}$$

which is sufficiently near the other value to justify the approximation.

It is outside the scope of the present work to discuss the design of fans and blowers. For information on these matters reference may be made to standard works on this subject. The following are mentioned for the guidance of the reader :—

The Fan, by C. H. Innes.
Mechanical Draft, by Sturtevant.
Boiler Draught, by H. K. Pratt.

Paper on "Design and Testing of Centrifugal Fans," by Heenan and Gilbert, *Proc. Inst. Civil Engs.*, Vol. CXXIII, p. 279.

Paper on "Turbo-Blowers and Compressors," by Guy and Jones. *Proc. South Wales Institute of Engineers*, January 18th, 1916.

Paper on "Centrifugal Blowers for High Pressures," by H. F. Schmidt. *Jour. Amer. Soc. Mech. Engs.*, Vol. 34.

The methods of testing fans and blowers are discussed in *The Testing of Motive Power Engines*, by R. Royds.

Chimneys.—The draught produced by an ordinary chimney is due to the difference of density between the gases inside the chimney and the air outside. For an adequate treatment of this subject reference may be made to *Chimney Design and Theory*, by W. W. Christie, and *Boiler Draught*, by H. K. Pratt.

There is one point, however, which deserves further mention, and that is, vertical boiler tubes carrying hot gases have a greater chimney action than with the same tubes set horizontally. For example, with vertical tubes 8 ft. long, mean temperature of gases in tubes 1400° F., and atmospheric temperature 60° F., and taking the gases in the tubes to be air at atmospheric pressure, 14·7 lb. sq. in. abs.

Average density inside tubes $=\dfrac{14{\cdot}7\times 144}{(1400+460)}\times\dfrac{1}{53{\cdot}2}$

$=\cdot 0214$ lb. per cub. ft.

" " of outside air $=\cdot 0214\times\dfrac{1860}{520}$

$=\cdot 0765$ lb. per cub. ft.

Pressure due to difference of density in column of 8 ft. $=(\cdot 0765-\cdot 0214)\times 8$

$=\cdot 441$ lb. per sq. ft.

or, $=\cdot 441\times\dfrac{1}{5{\cdot}2}=\cdot 085$ in. water.

Assuming that the gases would leave horizontal tubes at, say, 500° F. and enter the chimney at this temperature, the density in the first 8 ft. of chimney would be, at atmospheric pressure, $\cdot 0214\times\dfrac{1860}{(500+460)}=\cdot 0415$ lb. per cub. ft.

Corresponding pressure difference due to column of 8 ft. $=(\cdot 0765-\cdot 0415)\times 8$

$=\cdot 28$ lb. per sq. ft.

or, $=\dfrac{\cdot 28}{5{\cdot}2}=\cdot 054$ in. water.

Therefore, the vertical tubes would induce a draught of about $(\cdot 085 - \cdot 054) = \cdot 031$ in. water more than if the tubes were horizontal.

To Estimate the Dew Point Temperature for Flue Gases.— When a gas containing water vapour is cooled below the dew point some of the moisture in the gas condenses. Condensation sometimes occurs on the cold tube surfaces of a flue gas economiser, and as this is liable to cause corrosion it should be avoided whenever possible by arranging to have the feed water inlet temperature higher than the dew point of the flue gases. As an example consider the flue gases from an oil-fired furnace.

Flue gases per lb. oil 21 lb.; and water vapour 1·2 lb., or, $\frac{1\cdot 2}{21} = \cdot 057$ lb. per lb. gases, leaving $(1 - \cdot 057) = \cdot 943$ lb. non-condensable gases.

Let the total gas pressure $P_t = P_a + P_s = 2020$ lb. per sq. ft. where, P_a=pressure of non-condensable gases, and P_s that of the water vapour. If t° F. is the dew point temperature, and V the volume of ·943 lb. gases at the dew point, then, taking the gases to be the same density as air at the same pressure and temperature,

$$\frac{P_a V}{t+460} = 53\cdot 2 \times \cdot 943$$

But V=volume of ·057 lb. water vapour also, and taking V to be 15·5 cub. ft. for a trial calculation.

Then, $\frac{15\cdot 5}{\cdot 057} = 272$ cub. ft. per lb., and from steam tables $t = 109^\circ$ F., $P_s = 178$ lb. sq. ft., leaving $P_a = 1842$ lb. sq. ft.

Therefore, $V = \frac{\cdot 943 \times 53\cdot 2}{1842} \times 569 = 15\cdot 48$ cub. ft.

Thus, it may be assumed that $t = 109^\circ$ F., and the feed water inlet to the economiser in this case would not be below this temperature if condensation on the tubes is to be avoided.

EXAMPLES

To illustrate the application of the experimental data to certain problems in the transmission of heat the following

examples have been worked out for specified conditions of operation. It would be understood, however, that no claim is made that the calculations were absolutely precise at all points, and in any case, accurate calculation is hardly necessary because it would be impossible in practice to control the actual working conditions always in exact accordance with those specified.

EXAMPLE I.—The exhaust gases from a Diesel engine are required to raise steam in a boiler having 1-in. tubes through which the gases pass. An economiser is also to be used to heat the feed water, having 1-in. tubes through which the gases pass after leaving the boiler. The following particulars are specified : B.H.P. of Diesel engine 500 ; oil used per B.H.P. hour ·47 lb. ; exhaust gas temperature at engine 700° F. ; weight of gases per lb. of oil burned 30 lb., having a mean specific heat ·25.* Steam temperature 350° F. ; gas inlet at boiler 650° F. ; outlet from boiler 420° F. ; feed temperature to economiser 60° F., heated to 320° F. with counter current flow. Gas pressures above atmosphere.—At boiler inlet 6 in. water, at boiler outlet 3 in. water, and at economiser outlet atmospheric pressure.

To calculate the probable area of heating surface required, and the length and number of tubes required in both boiler and the economiser.

Boiler Section.

$$\text{Weight of gases per second} = w_1 = \frac{500 \times \cdot 47 \times 30}{3600} = 1 \cdot 96 \text{ lb.}$$

$$\text{Heat lost by gases per second} = 1 \cdot 96 \times \cdot 25\,(650 - 420)$$
$$= 113 \text{ B.Th.U.}$$

$$\text{Dry saturated steam per hour} = \frac{113 \times 3600}{900} = 452 \text{ lb., say, } 450 \text{ lb.}$$

For a first approximation equation 5, p. 29, may be used.

That is, $\dfrac{l}{m_1} = \text{M} \log_e \dfrac{\text{T}_1 - t_s}{\text{T}_2 - t_s}$

or, $\dfrac{l}{\frac{1}{48}} = \text{M} \log_e \dfrac{650 - 350}{420 - 350}$

and taking M to be 200 to 250, say, 225.

* The true mean specific heat would probably be a little greater than ·25.

Then, $l = \frac{1}{48} \times 225 \times \log_e 4{\cdot}3.$

$= 6{\cdot}83$ ft.

If ΔP = total drop of pressure through boiler, lb. per sq. ft.

δP = pressure drop due to resistance of tubes, lb. per sq. ft.

δP_1 = pressure drop to produce kinetic energy at inlet, lb. per sq. ft.

δP_0 = pressure loss at inlet.

Then, if there is no recovery at the tube outlet,

$\Delta P = \delta P + \delta P_1 + \delta P_0$ nearly.

Now, taking f, the coefficient of resistance, to be ·006, say,

Then, $\delta P = {\cdot}006 \times \frac{53{\cdot}2}{64{\cdot}4} \times \frac{\tau}{P} \times \frac{6{\cdot}83 \times 48}{1} \times \left(\frac{w_1}{a_1}\right)^2$, by equation 23, p. 85 of *Heat Transmission by Radiation, Conduction, and Convection.*

$$\tau = \frac{650+420}{2} + 460 = 995^\circ \text{ F. abs.}$$

$$P = 2120 \text{ lb. sq. in. abs.}$$

Thus, $\delta P = {\cdot}762\left(\frac{w_1}{a_1}\right)^2$ lb. sq. ft.

$$\delta P_1 = \frac{v^2{}_1 \rho_1}{64{\cdot}4} \text{ lb. sq. ft.}$$

$$= \frac{1}{64.4}\left(\frac{w_1}{a_1}\right)^2 \frac{1}{\rho_1} \text{ lb. sq. ft.}$$

$$\delta P_0 = \frac{0{\cdot}5}{64{\cdot}4}\left(\frac{w_1}{a_1}\right)^2 \frac{1}{\rho_1} \text{ lb. sq. ft., say}$$

Then, $\delta P_1 + \delta P_0 = \frac{1{\cdot}5}{64{\cdot}4\rho_1}\left(\frac{w_1}{a_1}\right)^2$ lb. sq. ft.

$$\frac{1}{\rho_1} = \frac{53{\cdot}2\tau_1}{P_1} = \frac{53{\cdot}2 \times (650+460)}{2120} \text{ cub. ft. per lb. (approx.).}$$

Then, $\delta P_1 + \delta P_0 = \frac{1{\cdot}5}{64{\cdot}4} \times 53{\cdot}2 \times \frac{1110}{2120} \times \left(\frac{w_1}{a_1}\right)^2$

$$= {\cdot}65\left(\frac{w_1}{a_1}\right)^2$$

$\therefore \quad \Delta P = (6-3)\ 5{\cdot}2$ lb. sq. ft.

$$= ({\cdot}762 + {\cdot}65)\left(\frac{w_1}{a_1}\right)^2$$

$$=1{\cdot}412\left(\frac{w_1}{a_1}\right)^2$$

or, $$\frac{w_1}{a_1}=\sqrt{\frac{3\times5{\cdot}2}{1{\cdot}412}}=3{\cdot}32 \text{ lb. per sec. per sq. ft.}$$

Since $w_1=1{\cdot}96$ lb. per sec.

then $a_1=\dfrac{1{\cdot}96}{3{\cdot}32}=\cdot591$ sq. ft.

Thus, number of tubes of 1 in. bore$=\dfrac{\cdot591}{\dfrac{\cdot785\times1^2}{144}}=108.$

Total heating surface $=108\times\dfrac{\pi}{12}\times6{\cdot}83$ sq. ft.

$=193$ sq. ft.

As a check, from Fig. 23 take $h=\cdot0037$ for $\dfrac{w_1}{a_1}=3{\cdot}32.$

Mean difference of temperature$=\dfrac{650-420}{\log_e\dfrac{650-350}{420-350}}=\dfrac{230}{\log_e 4{\cdot}3}$

$=158°$ F.

Then, $\cdot0037\times158\times193=113$ B.Th.U. per sec. which is the value calculated on p. 61.

Heat-flow per sq. ft. per hour $=\dfrac{113}{193}\times3600=2110$ B.Th.U.

Economiser Section with Counter Current Flow.

Heat supplied at economiser per second$=\dfrac{450}{3600}(320-60)$

$=32{\cdot}5$ B.Th.U.

Then, $1{\cdot}96\times\cdot25\times(420-t)=32{\cdot}5$

or, $t=354°$ F.

that is, the exit temperature of the gases is 354° F.

Now, mean temperature difference$=\dfrac{(420-320)-(354-60)}{\log_e\dfrac{100}{294}}$

$$=\frac{-194}{-1{\cdot}077}=180° \text{ F.}$$

If n=number of 1-in. tubes of length l ft.

$l\pi\frac{1}{12}n$=tube area, sq. ft.

Then, $h=\dfrac{32{\cdot}5\times 12}{180\times\pi\times l\times n}$ B.Th.U. sq. ft. sec. °F. diff.

Again, $\delta P=\cdot006\times\dfrac{53{\cdot}2}{64{\cdot}4}\times\dfrac{847}{2120}\times\dfrac{l}{\frac{1}{48}}\times\left(\dfrac{w_1}{a_1}\right)^2$

$=\cdot095\,l\left(\dfrac{w_1}{a_1}\right)^2$ lb. sq. ft.

$\delta P_1+\delta P_0=\dfrac{(1+\cdot5)}{64{\cdot}4\rho}\left(\dfrac{w_1}{a_1}\right)^2$ lb. sq. ft.

$=\dfrac{1{\cdot}5}{64{\cdot}4}\times 53{\cdot}2\times\dfrac{880}{2120}\times\left(\dfrac{w_1}{a_1}\right)^2$

$=\cdot515\left(\dfrac{w_1}{a_1}\right)^2$ lb. sq. ft.

$\Delta P=3\times 5{\cdot}2=15{\cdot}6=\left(\dfrac{w_1}{a_1}\right)^2\left\{\cdot095l+\cdot515\right\}$ lb. sq. ft.

Either $\dfrac{w_1}{a_1}$ could be chosen, or the length l determined approximately, as on p. 61.

If $\dfrac{w_1}{a_1}=4{\cdot}7$ is chosen to suit,

Then, $\dfrac{3\times 5{\cdot}2}{22{\cdot}1}=\cdot095l+\cdot515$

$l=\dfrac{\cdot19}{\cdot095}=2{\cdot}0$ ft.

Taking $h=\cdot0045$ at $\dfrac{w_1}{a_1}=4{\cdot}7$ from Fig. 23

$\cdot0045=\dfrac{32{\cdot}5\times 12}{180\times\pi\times 2\times n}$, or, $n=76{\cdot}7$, say, 77 tubes.

Total heating surface $=77\times\pi\times\frac{1}{12}\times 2=40{\cdot}3$ sq. ft.

Heat-flow per sq. ft. per hour $=\dfrac{32{\cdot}5\times 3600}{40{\cdot}3}$

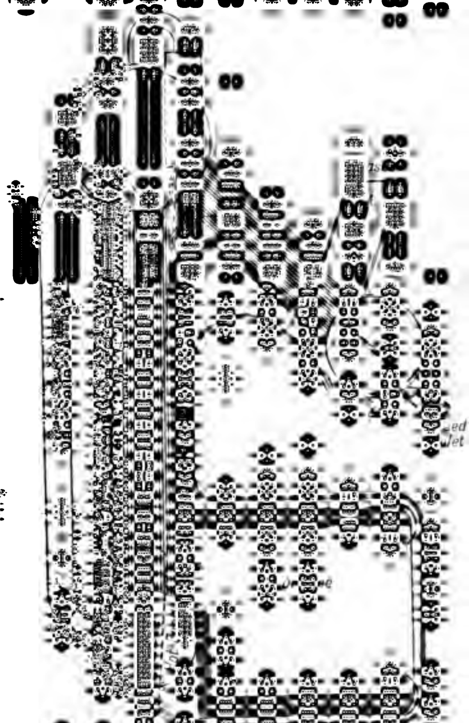
$=2900$ B.Th.U.

Retarders might be inserted in the boiler tubes to increase the rate of heat transmission and thereby reduce the necessary length of the tubes. The method of calculation would be the same as that adopted in Example IV. p. 82.

EXAMPLE II.—**High Speed Boiler with Flow of Gases along Tubes.**—Evaporation 50,000 lb. per hour dry saturated

(temperature
220° F., heated
this tempera-
F. with atmo-

% in excess of



of
ong

value of oil
l, carbon 86·5,

$\frac{3}{8}$ in. inside,

$\frac{1}{4}$ in. inside,

Chosen values of $\frac{w_1}{a_1} = \left\{\begin{matrix} \text{3 for evaporator} \\ \text{4 for economiser} \end{matrix}\right\}$ lb. gases per sec. per sq. ft. cross-section.

Gases flow along length of outside of tubes, with water inside the tubes flowing in the counter current direction. Induced draught by turbine-driven fan exhausting into feed heater.

The sketch arrangement of the proposed boiler in Fig. 32 is only intended to assist in fixing our ideas with respect to the following calculations and is not necessarily drawn to scale.

Air theoretically required per lb. oil for complete combustion $= \frac{100}{23}\left\{\frac{32}{12} \times \cdot 865 + 8 \times \cdot 128\right\} = 14 \cdot 5$ lb. nearly.

Air supplied per lb. oil $= 14 \cdot 5 \times 1 \cdot 4 = 20 \cdot 3$ lb.

Gases per lb. oil $= 20 \cdot 3 + 1 = 21 \cdot 3$ lb.

Composition of these gases by weight for complete combustion.

Carbon-dioxide (CO_2) $= \frac{44}{12} \times \cdot 865 = 3 \cdot 16$ lb.

Water vapour (H_2O) $= 9 \times \cdot 128 = 1 \cdot 15$ lb.

Nitrogen (N_2) $= \frac{77}{23} \times$ weight of oxygen used.

$$= \frac{77}{23}\left(\frac{32}{44} \times 3 \cdot 16 + \frac{8}{9} \times 1 \cdot 15\right)$$

$$= 11 \cdot 15 \text{ lb.}$$

Excess air $= 14 \cdot 5 \times \cdot 4 = 5 \cdot 8$ lb.

Percentage composition by weight

$$CO_2 = \frac{3 \cdot 16}{21 \cdot 3} \times 100 = 14 \cdot 9, \; H_2O = \frac{1 \cdot 15}{21 \cdot 3} \times 100 = 5 \cdot 4$$

$$N_2 = \frac{11 \cdot 15}{21 \cdot 3} \times 100 = 52 \cdot 4, \text{ Excess air} = \frac{5 \cdot 8}{21 \cdot 3} \times 100 = 27 \cdot 3.$$

Probable mean specific heat between furnace temperature and atmospheric temperature,

$CO_2 = \cdot 25$, $H_2O = \cdot 5$, $N_2 = \cdot 26$, air $\cdot 255$.

Mean specific heat of gases

$$= \cdot 25 \times \cdot 149 + \cdot 054 \times \cdot 5 + \cdot 524 \times \cdot 26 + \cdot 273 \times \cdot 255 = \cdot 27.$$

If T = furnace temperature, °F, then, neglecting all external

losses* and assuming complete combustion without any appreciable amount of dissociation.†

$$\cdot 27 \times 21\cdot 3 \times (T-70)=18350$$
$$T=3270^\circ \text{ F.}‡$$

where the air temperature near furnace is assumed to be about 70° F.

Heat required per lb. steam from water at 220° F.=1014 B.Th.U.

$$\text{Heat required per sec.}=\frac{50{,}000}{3600}\times 1014$$
$$=14100 \text{ B.Th.U.}$$

Probable boiler efficiency, allowing a loss of 5% by radiation, etc.,

$$=1-\frac{21\cdot 3\times \cdot 25\ (500-60)}{18350}-\cdot 05$$
$$=\cdot 822.$$

Then, Oil per sec. × 18,350 × ·822=14,100
or, Oil per sec.=·935 lb.
and, Gases per sec.=w_1=·935 × 21·3=19·9 lb.

Economiser Section.

$$\frac{w_1}{a_1}=4,\ \text{or,}\ a_1=\frac{w_1}{4}=\frac{19\cdot 9}{4}=4\cdot 97 \text{ sq. ft.}$$

h =·004 B.Th.U., sq. ft., sec., deg. Fahr. (from Fig. 23, p. 47).

Taking specific heat of gases ·25, and allowing 5% loss.

$$\text{Then, } 19\cdot 9\times \cdot 25\ (T-500)=\frac{50{,}000}{3600}\times(380-220)\times\frac{1}{\cdot 95}=\frac{2224}{\cdot 95}$$
$$\text{or,}\quad T=970^\circ \text{ F.}$$

* No doubt some heat would be lost externally by the furnace, the amount depending upon the furnace arrangements. If, for example, the furnace were surrounded by an air jacket through which the air passed on its way to the furnace, the net loss of heat would probably be comparatively small.

† Up to the highest temperatures here discussed the dissociation in ordinary furnaces is very small and practically negligible. The laws of dissociation are fully discussed in *Thermodynamics of Technical Gas Reactions* by F. Haber.

‡ Since the mean specific heat of furnace gases is only known approximately at high temperatures, the calculated temperature is uncertain to that extent. No allowance has therefore been made for any external loss of heat or for the small amount of dissociation mentioned previously. Most likely the true furnace temperature would be slightly below this calculated value.

$$\text{Mean difference of temperature} = \frac{(970-380)-(500-220)}{\log_e \dfrac{970-380}{500-220}} = 415^\circ\text{F}.$$

Then, $\cdot 004 \times 415 \times A_1 = 2224$

$$\text{or, } A_1 = \frac{2224}{\cdot 004 \times 415} = 1340 \text{ sq. ft. tube surface.}$$

$$\frac{l}{m_1} = \frac{A_1}{a_1} = \frac{1340}{4 \cdot 97} = 270 \text{ for heat-flow.}$$

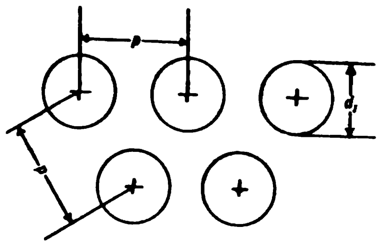


FIG. 33.

Referring to Fig. 33.

If n = number of tubes.
p = pitch of tubes.
d_1 = outside diameter of tubes.
a_1 = area for flow of gas.

Then, $a_1 = n\left\{\cdot 866\, p^2 - \frac{\pi}{4} d_1^2\right\}$

$$\text{Or,}\quad 4 \cdot 97 = \frac{n}{144}\left\{\cdot 866 \times \cdot 75^2 - \frac{\pi}{4} \times \cdot 5^2\right\}$$

$$n = \frac{4 \cdot 97 \times 144}{\cdot 292} = 2455$$

$$\text{But } A_1 = \pi d_1 l n = \frac{\pi \times \cdot 5 \times 2455 \times l}{12} = 1340$$

$$\therefore\quad l = \frac{1340 \times 12}{\pi \times \cdot 5 \times 2455} = 4 \cdot 2 \text{ ft.}$$

Assuming 16 rows of tubes,

Then, number of tubes per row $= \dfrac{2455}{16} = 153$

Width of each row $= 153 \times \cdot 75 = 115$ in.

Economiser Draught.

Let $\dfrac{l}{m_1} = 286$, say, allowing for surface of enclosing walls.

$P = 2000$ lb. sq. ft., say.

$$\tau = \frac{970+500}{2} + 460 = 1195^\circ \text{ F.}$$

$f^* = \cdot 02$.

* This value of f used in conjunction with the nominal dimensions allows for the thin permanent layer of soot which would adhere to the tubes.

Neglecting energy losses at inlet to the tubes and assuming kinetic energy of flow at outlet to be all lost,

Then, $\Delta P = \cdot 02 \times \frac{53 \cdot 2}{64 \cdot 4} \times 286 \times \frac{1195}{2000} \times 4^2 + \frac{1}{64 \cdot 4} \times 4^2 \times \frac{1}{\rho_1}$.

But $\frac{1}{\rho_1} = 53 \cdot 2 \times \frac{(970+460)}{2000} = 38$ cub. ft. per lb. nearly.

$\therefore \Delta P = 45 \cdot 2 + 9 \cdot 5 = 54 \cdot 7$ lb. sq. ft., or $10 \cdot 5$ in. water.

Evaporator Section.

Gas temperatures, inlet 3270° F, outlet 970° F.

$$\frac{w_1}{a_1} = 3, \; h = \cdot 0028, \text{ say.}$$

$$a_1 = \frac{w_1}{3} = \frac{19 \cdot 9}{3} = 6 \cdot 63 \text{ sq. ft.}$$

Heat to water per sec. $= \frac{50{,}000}{3600} \times 850 = 11{,}800$ B.Th.U.

$= 12{,}000$ B.Th.U., say.

Mean difference of temperature $= \dfrac{3270-970}{\log_e \dfrac{3270-406}{970-406}} = 1412°$ F.

Then, $\cdot 0028 \times 1412 \times A_1 = 12000$

$$\text{or, } A_1 = \frac{12000}{\cdot 0028 \times 1412} = 3040 \text{ sq. ft.}$$

$$\frac{l}{m_1} = \frac{A_1}{a_1} = \frac{3040}{6 \cdot 63} = 458 \text{ for heat-flow.}$$

$$a_1 = 6 \cdot 63 = \frac{n}{144} \left\{ \cdot 866 \times 1^2 - \frac{\pi}{4} \times \cdot 625^2 \right\}$$

$$\text{or, } n = \frac{6 \cdot 63 \times 144}{\cdot 56} = 1705 \text{ tubes.}$$

But $A_1 = 3040 = \pi \times \frac{\cdot 625}{12} \times 1705 \times l$.

or, $l = 10 \cdot 9$ ft.

If there are 15 rows of tubes,

Then, $\frac{1705}{15} = 114$ tubes per row.

And, width of row $= 114$ in.

Evaporator Draught.

Again,

$$\triangle P^* = \frac{\cdot 016 \times 53\cdot 2}{64\cdot 4} \times 470 \times \frac{(1412+406+460)}{2050} \times 3^2 + \frac{3^2}{64\cdot 4} \times \frac{1}{\rho_1}$$

where, $P = 2050$ lb. sq. ft., say.

$$\frac{l}{m_1} = 470$$

$$\frac{1}{\rho_1} = 53\cdot 2 \times \frac{(3270+460)}{2100} = 94\cdot 5 \text{ cub. ft. per lb. nearly.}$$

$$\therefore \quad \triangle P = 62\cdot 1 + 13\cdot 2 \qquad = 75\cdot 3 \text{ lb. per sq. ft.}$$

$$\text{or, } \frac{75\cdot 3}{5\cdot 2} = 14\cdot 5 \text{ in. of water.}$$

Total draught $= 10\cdot 5 + 14\cdot 5 = 25$ in. water, neglecting the small draught in the furnace.

At the induced draught fan the temperature of the gases would be 500° F. and the mean pressure about 2040 lb. per sq. ft.

$$\text{Volume gases per lb.} = 53\cdot 2 \times \frac{(500+460)}{2040} = 25 \text{ cub. ft. nearly.}$$

Volume per min. $= 25 \times 19\cdot 9 \times 60$ cub. ft.

Work† per min. on gases $= 25 \times 19\cdot 9 \times 60 \times 25 \times 5\cdot 2$ ft. lb.

$$\text{Horse power spent on gases} = \frac{25 \times 19\cdot 9 \times 60 \times 25 \times 5\cdot 2}{33{,}000}$$

$$= 117$$

Assuming a fan efficiency of $\cdot 6$

$$\text{Shaft horse power} \qquad = \frac{117}{\cdot 6} = 195.$$

An efficient steam turbine exhausting into a feed heater might use, say, 30 to 35 lb. of high-pressure steam per S.H.P. per hour, and taking a mean of 32·5 lb., and assuming each lb. of steam would give about 1000 B.Th.U. to the heater,

Then, Steam used by turbine per hour $= 195 \times 32\cdot 5$

$$= 6340 \text{ lb.}$$

* The coefficient of friction ·016 when used in conjunction with the nominal dimensions allows for the permanent thin layer of soot which would adhere to the tubes.

† This method of calculation is sufficiently accurate for the estimation of fan power.

If t=rise of temperature of feed water in heater,

$$6340 \times 1000 = 50{,}000 \times t$$

$$\text{or,} \quad t = \frac{6340 \times 1000}{50{,}000} = 127^\circ \text{ F.}$$

Thus, supposing the feed entering the heater had a temperature of 88° F., the feed temperature would be raised to 88+127=215° F. by the turbine exhaust.

Total steam per hour per sq. ft. evaporator heating surface = {16·45 lb. actual, or, 14·4 lb. from and at 212° F.

Total steam per hour per sq. ft. total heating surface = {11·4 lb. actual, or, 11·9 lb. from and at 212° F.

Available steam from boiler=50,000−6340
=43,660 lb. per hour.

Same Boiler Working at other Loads.—Having determined the boiler dimensions for the specified conditions in the previous calculations it is desirable to ascertain what the temperature conditions and boiler efficiencies are likely to be at other loads. As an example consider the boiler working at about half the previous load. For a first approximation assume the furnace temperature about 3100° F.; $\frac{w_1}{a_1}=2$ for economiser and 1·5 for evaporator; feed temperature to economise about 135° F. Same steam pressure 250 lb. sq. in. gauge.

Evaporator Section.

$$w_1 = \frac{19 \cdot 9}{2} = 9 \cdot 95 \text{ lb. per sec.}$$

If T_2° F.=temperature of gases leaving evaporator.

Heat lost by gases=$9 \cdot 95 \times \cdot 25 \times (3100 - T_2)$ B.Th.U. per sec. where the specific heat ·25 is taken at this low value to allow for external losses.

$$\text{Again, mean difference of temperature} = \frac{3100 - T_2}{\log_e \dfrac{3100-406}{T_2 - 406}}.$$

At $\frac{w_1}{a_1}=1 \cdot 5$, $h = \cdot 00185$ approximately.

Thus, heat gained by water=$\cdot 00185 \times 3040 \times$mean temp. diff. °F., B.Th.U. per sec.

By trial it was found that $T_2=690°$ F. suits these equations, and inserting this value, then,

$$9{\cdot}95\times{\cdot}25\times(3100-690) \qquad =6000 \text{ B.Th.U. per sec.}$$

and, $${\cdot}00185\times3040\times\frac{3100-690}{\log_e\dfrac{3100-406}{690-406}}=6010 \quad ,, \quad ,,$$

Taking the feed temperature to the evaporator to be about 280° F.

Then, $6000=W\times953$, where W = steam in lb. per sec.

$\therefore$ W = 6·29 lb. per sec.

or, W = 22,600 lb. of steam per hour.

Economiser Section.—Assumed feed temperature about 135° F. for a first approximation.

$$\frac{w_1}{a_1}=2, \text{ and } h={\cdot}0024 \text{ approximately.}$$

By trial it is found that the gases would leave at about 325° F. and the feed water leave at about 280° F.

Thus, mean difference of temperature $$=\frac{(690-280)-(325-135)}{\log_e\dfrac{690-280}{325-135}}$$

$$=286° \text{ F.}$$

Then, $${\cdot}0024\times1340\times286 \quad =919 \text{ B.Th.U. per sec.}$$

and, $$9{\cdot}95\times{\cdot}25\times(690-325) \quad =907 \quad ,, \quad ,,$$

Therefore, say, $910=6{\cdot}29\,(t_2-135)$

or, $t_2=280°$ F.

that is, the water leaves the economiser at 280° F.

The draught required is nearly proportional to $\left(\dfrac{w_1}{a_1}\right)^2$

Therefore, $$\triangle P=\frac{25}{4}=6{\cdot}25 \text{ in. of water nearly.}$$

Volume gases per lb. at fan $$=53{\cdot}2\times\frac{(325+460)}{2100}$$

$$=19{\cdot}9 \text{ cub. ft. nearly.}$$

Fan horse power $$=\frac{6{\cdot}25\times5{\cdot}2\times19{\cdot}9\times9{\cdot}95\times60}{33000}$$

$$=11{\cdot}7$$

and, taking the fan efficiency ·6,

Shaft horse power $=\frac{11 \cdot 7}{\cdot 6}=19 \cdot 5$.

With steam consumption at, say, 50 lb. per shaft horse power, and if each lb. of steam carries 1000 B.Th.U. to the feed heater, then,

Steam used per hour by turbine $=19 \cdot 5 \times 50=975$ lb. and, $975 \times 1000=22{,}600 \times (t_2-t_1)$

or, $t_2-t_1=43°$ F. nearly.

Assuming again that t_1 is about 88° F., the feed temperature (t_2) leaving the heater becomes,

$$t_2=88+43=131° \text{ F.}$$

Available steam from boiler $=22{,}600-975$
$=21{,}625$ lb. per hour.

The boiler efficiency at half load may be estimated in the following manner :—

Let x=oil burned per sec. in furnace, lb.
,, y=flue gases per lb. oil, lb.
,, z=boiler efficiency.

Then, $xy=9 \cdot 95$ lb. gases per sec.

$$z=\cdot 95-\frac{y \times \cdot 25 \times (325-60)}{18{,}350}=\cdot 95-\cdot 00361y.$$

where 5% is allowed for external losses.

Also, $x \times 18{,}350 \times z=\frac{22{,}600}{3600} \times 1097$ B.Th.U. per sec.

or, $\frac{9 \cdot 95}{y} \times 18350 \times (\cdot 95-\cdot 00361y)=\frac{22600 \times 1097}{3600}$.

$\therefore\ y=23$ lb.

This seems a reasonable sort of value compared with $21 \cdot 3$ lb. under the full load conditions.

Again, $x=\frac{9 \cdot 95}{23}=\cdot 432$ lb. oil per sec.

and, $z=\cdot 95-\cdot 00361 \times 23$
$=\cdot 867$.

Returning to the furnace conditions, where T is furnace temperature,

$$23 \times \cdot 27 \times (T-70)=18{,}350$$
$$\text{or,} \qquad T=3030° \text{ F.}$$

Originally it was assumed for a first approximation that the furnace temperature would probably be about 3100° F., which is seen to have been about 70° F. too high. If it were thought necessary these calculations might be modified by taking the furnace temperature to be about 3000° F. as a second and more accurate approximation. A little experience with calculations of the above nature soon enables one to adjust unknown factors in the calculations to suit the conditions.

It is further interesting to note in what manner the boiler dimensions have to be altered to cool the flue gases to various temperatures at the economiser outlet when evaporating a

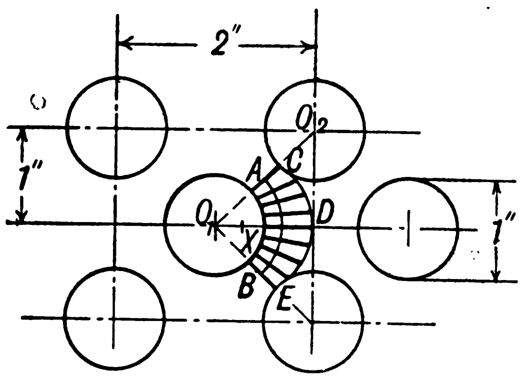


Fig. 34.—Mean area for flow of gases across tubes.

given weight of water. Calculations have been made by the author similar to those from pp. 67 to 71, but with the flue gases leaving the economiser at 400° F. and also at 600° F. For purpose of comparison the principal results are tabulated in Table 4.

Example III.—**High Speed Boiler having the Flow of Gas across the Tubes and the Water inside the Tubes.**

Particulars.—The particulars given for the boiler considered in Example II, p. 64, will suffice for the present purpose, except that $\frac{w_1}{a_1}=1 \cdot 5$ will be taken both for the evaporator and the economiser, and that the outside diameter of the tubes is 1 in., spaced as indicated in Fig. 34, having a length of tube 6 ft.

TABLE 4

BOILER CONDITIONS AND DIMENSIONS WHEN USING OIL-FIRED FIREBRICK FURNACE

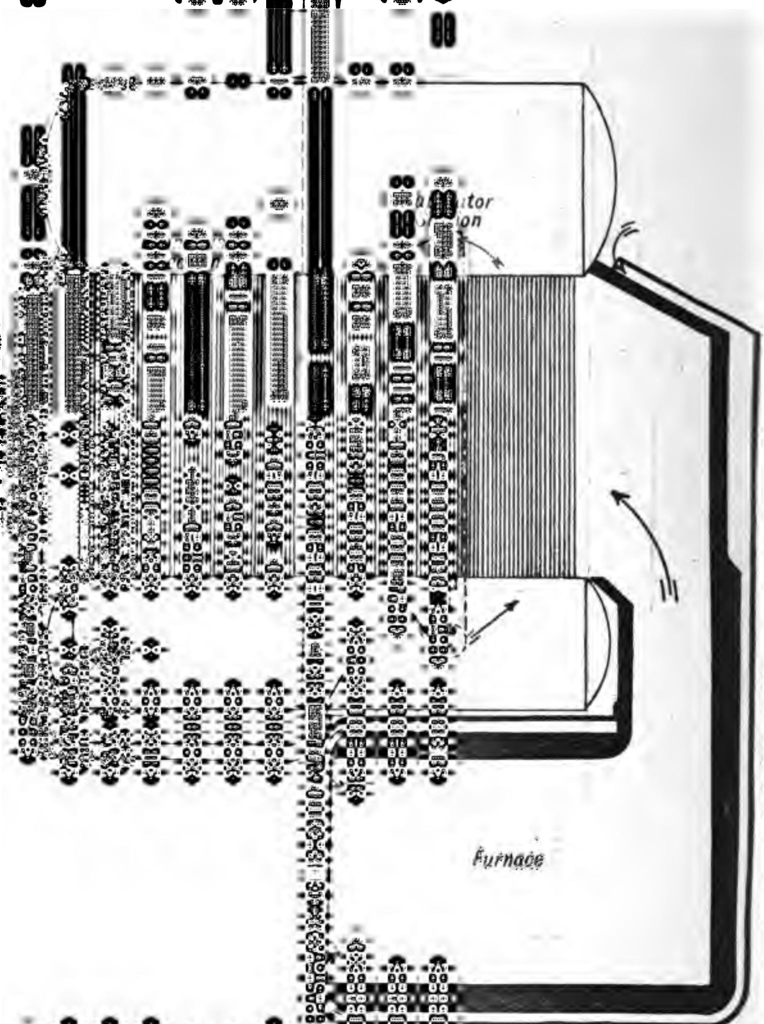
Total steam, per hour lbs.	Probable amount of steam used by turbine driving fan, lbs. per hour.	*Available steam from boiler, lbs. per hour.	Total Evaporation, lbs. per hour. Per sq. ft. evaporator. Actual.	Per sq. ft. evaporator. From and at 212° F.	Per sq. ft. evaporator and ecomiser. Actual.	Per sq. ft. evaporator and ecomiser. From and at 212° F.	Weight of gases per lb. oil burned lbs. (assumed).	Gas Temperatures °F. Evaporator. Inlet.	Evaporator. Outlet.	Economiser. Outlet.
50,000	6340	43,660	15·5	13·6	10·0	10·5	21·3	3270	885	400
,,	6340	43,660	16·45	14·4	11·4	11·9	,,	,,	970	500
,,	6810	43,190	17·4	15·2	12·7	13·3	,,	,,	1060	600

Water temperatures at economiser °F. Inlet.	Water temperatures at economiser °F. Outlet.	Steam press, lbs. sq. in. gauge.	Steam temperature °F.	Evaporator Tubes. Diameter, inches. Outside.	Evaporator Tubes. Diameter, inches. Inside.	Evaporator Tubes. Length feet.	Evaporator Tubes. Number.	Economiser Tubes. Diameter, inches. Outside.	Economiser Tubes. Diameter, inches. Inside.	Economiser Tubes. Length feet.	Economiser Tubes. Number.
220	380	250	406	$\frac{5}{8}$	$\frac{3}{8}$	11·9	1651	$\frac{1}{2}$	$\frac{1}{4}$	5·7	2380
,,	,,	,,	,,	,,	,,	10·9	1705	,,	,,	4·2	2455
,,	,,	,,	,,	,,	,,	10·0	1754	,,	,,	3·3	2520

Heating Surface. Evaporator. sq. ft.	Heating Surface. Economiser. sq. ft.	Heating Surface. Total. sq. ft.	Probable boiler efficiency overall.	Total draught, inches water.	* Probable shaft horse power at fan (induced draught).	Values of $\frac{w_1}{a_1}$ Evaporator.	Values of $\frac{w_1}{a_1}$ Economiser.	Velocity of feed water at inlet to economiser tubes ft. per sec.
3220	1770	4990	·851	28	195	3	4	·29
3040	1340	4380	·822	25	195	,,	,,	·28
2870	1080	3950	·794	23	210	,,	,,	·27

* With forced draught the power required by the fan would only be about one-half these values, being, in fact, nearly proportional to the absolute volume, and therefore to the absolute temperature, of the gases passing through the fan.

…rs across the tubes the …he flow of the gases, is



…of high-speed boiler with …bes.

…e stated approximately. …author usually estimates

the average value of a_1 in the manner shown in Fig. 34. With radius O_1A of one-half the diagonal pitch O_1O_2, the arc AB is drawn, and also from a point X the arc CDE is drawn. The arc AB is then divided into a suitable number of equal parts and radial lines drawn from O_1. The average width between the tube and the arc CDE along these radial lines is then estimated and this, when multiplied by length of tube, is taken to represent approximately the area available for the flow of the gas round this side of one tube. For this particular case the area per tube becomes $(\cdot\frac{88}{12}\times 6)=\cdot 44$ sq. ft.

The sketch arrangement shown in Fig. 35 is intended to illustrate one possible design for the conditions here specified. Other arrangements are possible; for example, the boiler shown in Fig. 32 might be arranged with baffles to direct the gas-flow backwards and forwards across the tubes, instead of allowing the gases to flow lengthwise along the tubes.

Economiser Section.—The calculations relating to the economiser are complicated by the rise of temperature of the water which necessarily occurs along the length of each tube. With the feed water inlet 220° F. and the mean outlet temperature 380° F. it will be taken, for a first approximation, that the mean water temperatures are 320° F. at the gas inlet and 270° F. at the gas outlet. The mean temperature difference can be approximately estimated by the usual expression.

$$\text{Mean difference}=\frac{(970-320)-(500-270)}{\log_e \dfrac{970-320}{500-270}}$$

$$=403^\circ \text{ F.}$$

Taking $h=\cdot 0027$.

Then, $\cdot 0027\times 403\times A_1=2224$

or, $A_1=2040$ sq. ft.

Since $\dfrac{w_1}{a_1}=1\cdot 5$,

and $w_1=19\cdot 9$ (as in Example II).

Then, $a_1\doteq\dfrac{19\cdot 9}{1\cdot 5}=13\cdot 25$ sq. ft.

With n tubes per row and length of tube 6 ft.,

$$(n-\cdot 5)\times\cdot 44=13\cdot 25$$

$$\text{or,}\quad n=\frac{13\cdot 25}{\cdot 44}+\cdot 5=30\cdot 6,\ \text{say, } 31.$$

The ·5 allows for one tube in each row being placed close to the enclosing wall. If the two outer tubes on each side are brought into line to form the enclosing walls it would be necessary to add two extra tubes in each row for this purpose.

$$\text{No. of rows}\times 31\times 6\times\pi\times\tfrac{1}{12}=2040$$
$$\text{or, No. of rows}=42.$$
$$\text{Total number of tubes}=31\times 42=1300.$$

With the spacing shown in Fig. 34 the overall length of the economiser section of tubes would be about 3 ft. 6 in.

The following method of calculation can be used to check the mean temperatures of the water assumed at the outset. Thus, taking the weight of feed water passing through each tube to be the same.

$$\text{Water per tube per sec.}=\frac{50{,}000}{3600}\times\frac{1}{1300}=\cdot 0107\ \text{lb.}$$

$$\text{Surface of each tube}=6\times\frac{\pi}{12}=1\cdot 57\ \text{sq. ft.}$$

Then, at gas inlet end,

$$\cdot 0027\times(970-320)\times 1\cdot 57=\cdot 0107\times\text{temp. rise,}$$
$$\text{or, Rise of temp.}=257^\circ\ \text{F.}$$

Therefore, temperature at top of tube $=220+257=477^\circ$ F.

But the steam temperature being 406° F., the temperature of the water would not exceed 406° F., and therefore a certain amount of steam would be generated in the first few rows of tubes, being afterwards condensed by the colder water from the other tubes. The average of 477° F. and 220° F. is 348·5 °F. and in view of the above remarks a mean water temperature of 320° F. is a likely value for the first row of tubes.

Similarly, at the gas outlet end,

$$\cdot 0027\times(500-270)\times 1\cdot 57=\cdot 0107\times\text{temp. rise,}$$
$$\text{or, Temp. rise}=91^\circ\ \text{F.}$$

Mean temperature of water $=220+\frac{91}{2}=266$ °F.

As it is hardly likely that each tube would carry exactly

the same amount of feed water this value is sufficiently near the 270° F. originally assumed to justify the original estimate.

Evaporator Section.—With $\frac{w_1}{a_1}=1{\cdot}5$, the value of h may be taken ·0028, and the calculations are practically the same as for Example II, p. 69.

Thus, mean difference of temperature$=1412°$ F.

$$\cdot 0028\times 1412\times A_1=12{,}000$$

$$\text{or,}\quad A_1=3040 \text{ sq. ft.}$$

With length of tube 6 ft., and $a_1=13{\cdot}25$ sq. ft.

No. of tubes per row $=31$

No. of rows $=\dfrac{3040}{31\times\frac{6}{12}\times\pi}=62{\cdot}5$, say, 63.

No. of tubes $=31\times 63=1950$

and,

Length of tube section $=5$ ft. 3 in.

Economiser Draught.—$\dfrac{l}{m_1}=\dfrac{A_1}{a_1}=\dfrac{2040}{13{\cdot}25}=\left.\begin{array}{l}154, \text{ or, say, } 160\\ \text{with enclosing walls}\end{array}\right\}$

Taking $f=\cdot 045$.

$$\Delta P=\cdot 045\times\frac{53{\cdot}2}{64{\cdot}4}\times 160\times\frac{1195}{2100}\times 1{\cdot}5^2+\frac{36}{64{\cdot}4}\times 1{\cdot}5^2$$

where, 1195° F. is mean absolute temperature,
2100 lb. sq. ft. is mean absolute pressure,
36 is the approximate volume in cub. ft. per lb. gases at inlet to economiser.

$$\therefore\ \Delta P=7{\cdot}61+1{\cdot}26=8{\cdot}9 \text{ lb. sq. ft.}$$

$$\text{or, } \frac{8{\cdot}9}{5{\cdot}2}=1{\cdot}71 \text{ in. water.}$$

Evaporator Draught.—$\dfrac{l}{m_1}=\dfrac{3040}{13{\cdot}25}=229{\cdot}5$, or, say, 240 with enclosing walls.

With $f=\cdot 045$

$$\Delta P=\cdot 045\times\frac{53{\cdot}2}{64{\cdot}4}\times 240\times\frac{2278}{2100}\times 1{\cdot}5^2+\frac{94{\cdot}5}{64{\cdot}4}\times 1{\cdot}5^2$$

$$=21{\cdot}8+3{\cdot}3=25{\cdot}1 \text{ lb. sq. ft.}$$

$$\text{or, } \frac{25{\cdot}1}{5{\cdot}2}=4{\cdot}83 \text{ in. water.}$$

Total draught$=1{\cdot}71+4{\cdot}83=6{\cdot}54$ in., or, say, 7 in. water.

Fan Power.—At 500° F. the volume per lb. gases is about 25 cub. ft. and with induced draught,

$$\text{Horse power}=\frac{7\times 5{\cdot}2\times 19{\cdot}9\times 25\times 60}{33{,}000}=32{\cdot}9$$

and, assuming as fan efficiency ·6,

$$\text{Shaft horse power}=\frac{32{\cdot}9}{{\cdot}6}=55.$$

Taking 35 lb. of steam used per shaft horse power per hour by the fan turbine.

Then, steam used per hour$=35\times 55=1925$ lb.

$$\text{Rise of temperature at feed heater}=\frac{1925\times 1000}{50{,}000}=38{\cdot}5^\circ \text{ F.}$$

If the feed to the heater had the temperature of the air pump discharge, say, 88° F., the temperature at the outlet from the heater would be $88+38{\cdot}5=126{\cdot}5^\circ$ F. Originally the feed temperature at the economiser inlet was taken at 220° F., and if no other auxiliary steam were available for heating the feed the calculations could be modified to suit the feed temperature 126·5° F.

The rate of heat transmission in the boilers shown in Figs. 32 and 35 might be increased by inserting strips of metal among the tubes in the manner represented in Fig. 36. The resistance offered to the flow of the gases would, of course, be increased also, and, unless the introduction of such strips of metal was justified by the probable reduction of the space occupied by the boiler, their use hardly seems desirable.

Example IV.—This example refers to a boiler of the locomotive type, working at a high rate of evaporation, under the following specified conditions :—

Evaporation per hour 20,000 lb. of dry saturated steam at 200 lb. per sq. in. gauge (temperature 388° F.); feed temperature (from feed heater) 200° F.; atmospheric temperature 60° F.; coal fired per sq. ft. of grate per hour 120 lb.; efficiency of boiler 54% with a smoke-box temperature of 600° F.; lower calorific value of coal 13,500 B.Th.U. per lb.; gases per lb. of coal 11 lb., $\frac{w_1}{a_1}=2{\cdot}5$.

Case 1—Plain tubes 1·75 in. outside and 1·53 in. inside diameter, with smoke-box temperatures; A 600° F.[1], B 700° F., and C 800° F.

Case 2—With straight retarders inserted in the same diameter tubes, and with the same smoke-box temperatures, referred to as D, E, and F respectively.

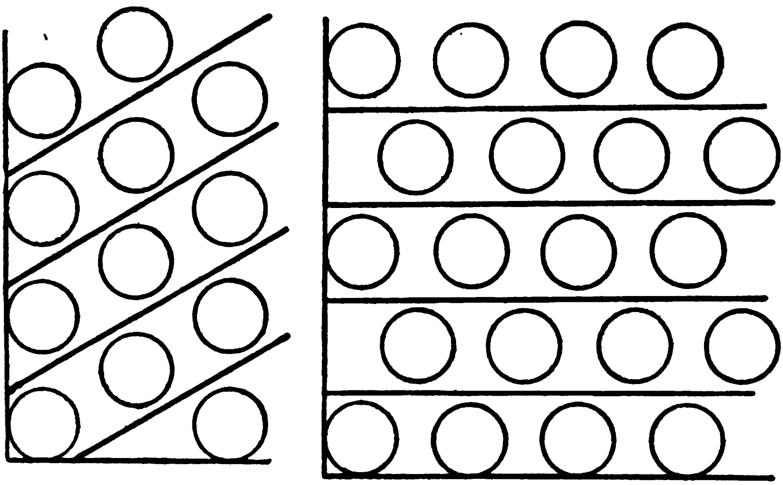


FIG. 36.—Radiating surfaces among boiler tubes.

Case 1, with smoke-box temperature 600° F.

$$\text{Coal fired per hour} = \frac{1031 \times 20{,}000}{\cdot 54 \times 13{,}500} = 2830 \text{ lb.}$$

$$\text{Area of grate} = \frac{2830}{120} = 23{\cdot}6 \text{ sq. ft.}$$

$$\text{Gas per second} = w_1 = \frac{2830}{3600} \times 11 = 8{\cdot}65 \text{ lb.}$$

$$a_1 = \frac{8{\cdot}65}{2{\cdot}5} = 3{\cdot}46 \text{ sq. ft.}$$

If n=number of tubes.

$$\text{Then, } \frac{n \times \cdot 785 \times 1{\cdot}53^2}{144} = 3{\cdot}46,$$

or, n=270 tubes.

Let T_1=furnace temperature, °F.

Then, according to the method of calculation discussed on p. 25,

$$23{\cdot}6\times\frac{1600}{3600}\left\{\left(\frac{T_1+460}{1000}\right)^4-\left(\frac{400+460}{1000}\right)^4\right\}+8{\cdot}65\times{\cdot}25\times\left(T_1-600\right)=\frac{1031\times20{,}000}{3600}$$

$$=5730\text{ B.Th.U.}$$

By trial it was found that $T_1=2750°$ F. solves this equation, and thus,

$$\text{Mean difference of temperature}=\frac{2750-600}{\log_e\dfrac{2750-388}{600-388}}=892°\text{ F.}$$

Taking $h={\cdot}003$, and letting A=area of tube-heating surface,

$$\text{Then, } {\cdot}003\times892\times A=8{\cdot}65\times{\cdot}25\times(2750-600),$$

$$\text{or, } A=1740\text{ sq. ft.}$$

If l=length of tubes, ft.

$$\frac{270\times\pi\times1{\cdot}53\times l}{12}=1740$$

$$\text{or, } l=16{\cdot}1\text{ ft.}$$

Similar calculations were made with the smoke-box temperatures 700° F. and 800° F., and the whole of the results are given in lines A, B, and C in Table 5.

In Table 5 the lines D, E, and F give the estimated values when using straight retarders, and these are respectively comparable to lines A, B, and C. To obtain these values, use was made of equations 1, 2, and 3 of p. 196* to estimate the rate of heat transmission at the two ends of the tube, and the average was taken. A reference to column 9 of Table 22* shows that this calculated value could be increased by at least 15%. Thus in example D the radiation from the retarder was estimated to increase the value of h from ·003 to ·0038. Increasing this by 15% of ·003 gives $\cdot0038+\cdot15\times\cdot003=\cdot0043$. The calculation of the heating surface and length of tubes then proceeds as in the detailed example above.

The results of another example is given in Table 5 referring to a boiler working at a low rate of evaporation, under the following conditions:—

Evaporation per hour 5000 lb. of dry saturated steam at

* *Heat Transmission by Radiation, Conduction, and Convection.*

TABLE 5.

CHARACTER OF BOILER.	Reference.	$\frac{w_1}{a_1}$	Steam pressure lbs. sq. in. gauge.	Temp. of smoke-box gases. °Fah.	Number of tubes.	Length of tubes.	Heating surface of tubes. sq. ft.	Reduction of tube-heating surface due to straight retarders.		Nett reduction of surface after adding area of retarders.		Reduction in length of boiler due to straight retarders.	Probable boiler efficiency.	REMARKS.
								Sq. ft.	Percentage of Col. 7.	Sq. ft.	Percentage of Col. 7.	Ft.	per cent.	
Boiler evaporating	A	2·5	200	600	270	16·1	1740	—	—	—	—	—	54	No retarders in use.
20,000 lb. of steam	B	,,	,,	700	281	13·5	1520	—	—	—	—	—	52	
per hour.	C	,,	,,	800	292	11·6	1360	—	—	—	—	—	50	
Tubes 1·75 in. outside diam. and 1·53 in.		,,	,,	600	281	10·7	1210	530	30·5 of 7a	154	8·9 of 7a	5·4	54	Straight retarders in tubes.
inside diam.		,,	,,	700	292	9·1	1060	460	30·2 of 7b	127	8·4 of 7b	4·4	52	
		,,	,,	800	304	7·8	948	412	30·3 of 7c	116	8·5 of 7c	3·8	50	
Boiler evaporating	G	0·7	160	500	344	12·4	1708	—	—	—	—	—	74	No retarders in use.
5000 lb. of steam per	H	,,	,,	600	364	9·6	1394	—	—	—	—	—	70	
hour.	K	,,	,,	700	384	7·8	1203	—	—	—	—	—	66	
Tubes 1·75 in. outside diam. and 1·53 in.	L	,,	,,	500	359	7·3	1055	653	38·2 of 7g	326	19·1 of 7g	5·1	74	Straight retarders in tubes.
inside diam.	M	,,	,,	600	379	5·7	862	532	38·1 of 7h	263	18·9 of 7h	3·9	70	
	N	,,	,,	700	401	4·6	745	458	38·0 of 7k	226	18·8 of 7k	3·2	66	
	1	2	3	4	5	6	7	8	9	10	11	12	13	

160 lb. per sq. in. gauge (temperature 371° F.); feed temperature 120° F.; atmospheric temperature 60° F.; coal fired per sq. ft. of grate per hour 25 lb.; efficiency of boiler 74%, with a smoke-box temperature of 500° F.; lower calorific value of coal 13,500 B.Th.U. per lb.; gases per lb. of coal fired 20 lb.; $\frac{w_1}{a_1} = \cdot 7$.

Case 3—With tubes 1·75 in. outside and 1·53 in. inside diameter, with smoke-box temperatures G, 500° F.; H, 600° F.; K, 700° F. Gases pass direct from the fire-box to the tubes.

Case 4—With straight retarders inserted in the same diameter tubes, and with the same smoke-box temperatures, referred to as L, M, and N respectively. Gases pass direct from the fire-box to the tubes.

The calculations for these cases were made in a similar manner to those relating to the previous example, and the results are given in Table 5 in the lines G to N.

It would be noted from columns 8 to 11 that the straight retarders are estimated to cause a considerable reduction of the heating surface, and in column 12 there is given the corresponding reduction in the length of the boiler.

Looking at these calculated values in another way, a comparison between the lines C and D of Table 5 indicates that, with the length of tube of about 11 ft., the introduction of straight retarders in the tubes might be expected to raise the efficiency of the boiler from the value 50% to 54% under these particular conditions of operation. Similarly from lines K and L it might be inferred that the efficiency for these conditions of operation would be improved from the value 66% to 74%.

The resistance offered to the flow of the gases through the plain tubes can be estimated by means of equation 2, p. 53, selecting values of f from Fig. 28, p. 53, if the pressure required to create the kinetic energy of flow is estimated separately, or from Fig. 27, and eqn. 1, if this pressure drop is included in the value of f. For the retarders the resistance can be estimated from the results given in Fig. 95, p. 198.* The furnace draught would depend to some extent upon the thick-

* *Heat Transmission by Radiation, Conduction, and Convection*

ness of the bed of fuel, and on the size and character of the coal. Some idea of its magnitude can be obtained from the data shown in Figs. 10, 11, and 18.

EXAMPLE IV (*continued*).—The influence of a deposit of scale on the transmission of heat through a boiler plate has been discussed to some extent on pp. 56 and 109,* and it was seen that unless the resistance due to the scale was very high it had comparatively little influence on the rate of heat transmission from the gases to the water in a boiler. If the thickness of a scale and its conductivity are known it is possible to estimate the probable influence on the efficiency of a boiler. To illustrate the method of calculation and the character of the results obtained, the two boilers A and G in Table 5 will be taken to have an additional deposit of scale $\frac{3}{16}$ in. thick, having a conductivity ·0005 B.Th.U. per sq. ft. per sec. per deg. Fahr. per ft. of thickness.

For boiler A the original value of the rate of heat transmission, h, was ·003. The new value, h', with the $\frac{3}{16}$-in. scale on the tubes will be,

$$\frac{1}{h'}=\frac{1}{h}+\frac{1}{\dfrac{\cdot 0005}{\frac{3}{16}\times\frac{1}{12}}}=\frac{1}{\cdot 003}+\frac{1}{\cdot 032}$$

or, $h'=\cdot 00275$

If t_m is the mean difference of temperature, and T_2 the outlet gas temperature, then,

$$\cdot 00275\times t_m\times 1740=8\cdot 65\times\cdot 25\times(2750-T_2)\text{ nearly.}$$

To solve this, either t_m or T_2 would have to be assumed and the other calculated and then the values adjusted until the equations are satisfied, for,

$$t_m=\frac{2750-T_2}{\log_e\dfrac{2750-388}{T_2-388}}\ {}^\circ\text{F}.$$

A few trial calculations soon gives the required values ; thus it is found that $t_m=950^\circ$ F. and $T_2=650^\circ$ F. nearly.

Without the scale on the tubes the percentage loss of heat by the flue gases was,

$$\frac{11\times\cdot 25\times(600-60)\times 100}{13{,}500}=11\%\text{ nearly.}$$

* *Heat Transmission by Radiation, Conduction, and Convection.*

With the $\frac{3}{16}$-in. scale the corresponding loss of heat is,

$$\frac{11\times\cdot25\times(650-60)\times100}{13,500}=12\%\text{ nearly.}$$

In this case, then, the efficiency of the boiler is only reduced by $(12-11)=1\%$ due to the scale. Similar calculations for boiler G shows that the loss of heat by the flue gases is only increased from about 16·3% to about 17%, thus only causing a loss of efficiency of $(17-16\cdot3)=0\cdot7\%$.

From these calculations it would therefore appear that whenever a boiler shows an appreciable loss of efficiency due to scale on the tubes, either the scale is very thick, or if thin, it is practically non-porous and in very bad thermal contact with the tube or plate. This would also signify that the temperature of the metal would be correspondingly raised, possibly to a dangerous extent.

EXAMPLE V.—**Steam Superheater.**—The following conditions are specified :—

Steam per hour 10,000 lb., superheated from the saturation temperature 370° F. to 550° F. at constant pressure.

Dryness fraction of steam at inlet ·99. Specific heat of superheated steam ·5.*

Gas temperatures, inlet 1000° F., outlet 600° F.; specific heat of gases taken at ·25.

Steam flows through tubes of 1-in. bore, with outside diameter 1·2 in. Gas flows along the length of the tube on the outside in the counter current direction.

If w_g=weight of gas-flow, lb. per sec.

a_g=area of section for gas-flow, sq. ft.

w_s and a_s the respective values for the steam-flow.

As an example of the method of calculation,

Let $\frac{w_g}{a_g}=2$, and $\frac{w_s}{a_s}=5$.

$$h_g=\cdot0025 \text{ and } h_s\dagger=\frac{\cdot5}{\cdot25}\times\cdot0045=\cdot009.$$

* The specific heat of superheated steam varies to some extent with the temperature and pressure (see Marks and Davis' steam tables), but as it is always difficult to estimate exactly the amount of heat to be supplied because of uncertainty of the quality of the saturated steam, the value ·5 is considered sufficiently exact for the purposes of calculation.

† For a given value of $\frac{w}{a}$ the value of h is approximately proportional to the specific heat of the gas.

Heat required per second to dry steam

$$=\frac{10{,}000}{3600}\times 853\times(1-\cdot 99)$$
$$=23\cdot 7 \text{ B.Th.U.}$$

Heat required per second for superheating

$$=\frac{10{,}000}{3600}\times \cdot 5\times(550-370)$$
$$=250 \text{ B.Th.U.}$$

$$\therefore \quad w_g\times\cdot 25\times(1000-600) =250+23\cdot 7$$
$$\text{or,} \quad w_g=2\cdot 74 \text{ lb. per sec.}$$

$$a_g=\frac{w_g}{2}=\frac{2\cdot 74}{2}=1\cdot 37 \text{ sq. ft.}$$

$$w_s=\frac{10{,}000}{3600}=2\cdot 78 \text{ lb. per sec.}$$

and, $$a_s=\frac{w_s}{5}=\frac{2\cdot 78}{5}=\cdot 556 \text{ sq. ft.}$$

$$\text{No. of tubes}=n=\frac{\cdot 556}{\cdot 785\times\frac{1}{144}}=102.$$

Drying the Steam.—The value of h_s under these conditions is not known, and is probably greater in value than when superheating; but as the heat required to dry the steam is generally small compared with that for superheating it may be taken to be the same as for superheating without sensible error.

Neglecting the small amount of conduction along the tube and the small drop of temperature through the metal, then, at the steam inlet, where tube temperature is θ,

$$h_g\,(600-\theta)\,1\cdot 2=h_s\,(\theta-370)\times 1$$

$$\text{or,} \quad \theta=\frac{h_g\times 600\times 1\cdot 2+h_s\times 370\times 1}{h_g\times 1\cdot 2+h_s}$$

$$=\frac{\cdot 0025\times 600\times 1\cdot 2+\cdot 009\times 370}{\cdot 0025\times 1\cdot 2+\cdot 009}$$

$$=427^\circ \text{ F.}$$

Again, if T_g=gas temperature at end of drying process.*

Then, $2\cdot 78\times 853\times(1-\cdot 99)=23\cdot 7=2\cdot 74\times\cdot 25\times(T_g-600)$

$$\text{or,} \quad T_g=635^\circ \text{ F.}$$

* It is likely, however, that no definite line of demarcation exists between the saturated and the superheated steam, as there may be differences of steam condition across any section of a tube.

and at this point, $\cdot 0025 \times 1 \cdot 2\ (635-\theta)=\cdot 009 \times (\theta-370) \times 1$

or, $\theta=436^\circ$ F.

The probable mean difference of temperature t_m between the gases and the tube during drying of the steam would be

$$t_m=\frac{(635-436)-(600-427)}{\log_e \frac{635-436}{600-427}}=186^\circ \text{ F.}$$

If A_g=square feet of heating surface on gas side,

Then, $\cdot 0025 \times 186 \times A_g=23\cdot 7$

or, $A_g=51$ sq. ft.

$$=\pi \times \frac{1\cdot 2}{12} \times l \times 102$$

or, $l=\frac{51 \times 12}{\pi \times 1\cdot 2 \times 102}=1\cdot 6$ ft. for drying the steam.

Superheating the Steam.

At gas inlet $\cdot 0025 \times 1\cdot 2 \times (1000-\theta)=\cdot 009 \times 1 \times (\theta-550)$

or, $\theta=662^\circ$ F.

Probable mean difference of temperature t_m between gases and tube during superheating,

$$t_m=\frac{(1000-662)-(635-436)}{\log_e \frac{1000-662}{635-436}}$$

$$=262^\circ \text{ F.}$$

$\therefore\ \cdot 0025 \times 262 \times A_g=250$B.Th.U. per sec.

or, $A_g=382$ sq. ft.

and, $l \times \frac{\pi \times 1\cdot 2}{12} \times 102=382$

or, $l=11\cdot 9$ ft.

Then, total length of each tube$=11\cdot 9+1\cdot 6$

$=13\cdot 5$ ft.

Similar calculations have been made with $\frac{w_s}{a_s}=10$ and 15, with $\frac{w_g}{a_g}=2$. Also, with $\frac{w_s}{a_s}=5$, 10, and 15, when $\frac{w_g}{a_g}=1$. The principal results are tabulated in Table 6. Thus it is seen that with $\frac{w_g}{a_g}=2$ increasing the value of $\frac{w_s}{a_s}$ from 5 to 15

only reduces the total heating surface from about 433 to about 365 sq. ft. Similarly with $\frac{w_g}{a_g}=1$ the heating surface is only reduced from about 650 to about 582 sq. ft. for the same change of $\frac{w_s}{a_s}$. It would be noted, however, that an increase of $\frac{w_g}{a_g}$ from the value 1 to 2 causes a reduction of surface of about one-third.

TABLE 6

$\frac{w_g}{a_g}$	$\frac{w_s}{a_s}$	Steam Temperatures. Inlet. °F.	Steam Temperatures. Out'et. °F.	Gas Temperatures. Inlet. °F.	Gas Temperatures. Outlet. °F.	Calculated Tube Temperatures. At Steam Inlet. °F.	Calculated Tube Temperatures. Highest Value. °F.	Outside Heating Surface, sq. ft.	Length of each Tube, feet.	No. of Tubes.
2	5	370	550	1000	600	427	662	433	13·5	102
2	10	,,	,,	,,	,,	406	622	385	24·0	51
2	15	,,	,,	,,	,,	397	600	365	34·1	34
1	5	,,	,,	,,	,,	·409	625	650	20·3	102
1	10	,,	,,	,,	,,	393	596	600	37·4	51
1	15	,,	,,	,,	,,	387	582	582	54·4	34

Although the rate of steam-flow through the superheater tubes does not greatly influence the required superheating surface it may have an important influence on the temperature of the tubes, as is indicated by the calculated values given in Table 6. With separately fired superheaters, and in some types of fire-tube superheaters, the gases usually have a high temperature and there is some danger of overheating the tubes.

The following results given in Table 7 have been calculated in the same manner as is shown on p. 88, but having the gas temperature at the inlet 2000° F. and at outlet 700° F., using counter current flow, with $\frac{w_g}{a_g}=2\cdot5$, the steam flowing through 1-in. bore tubes and the gases outside. The steam temperatures are the same as those specified on p. 86.

It is seen from this table that altering the rate of steam-flow from $\frac{w_s}{a_s}=25$ to 100 only reduces the calculated heating surface from about 130 sq. ft. to about 121 sq. ft., but the calculated

tube temperature at the steam outlet (gas inlet) is reduced by about 100° F.

It would be observed, however, that under ordinary conditions it might be impracticable to provide for counter current flow with tubes of the lengths given in Tables 6 and 7. In locomotive superheaters, for instance, having the superheater tubes inserted into the fire tubes from the smoke-box end, either two or four steam passes are adopted for each tube element. Thus with two passes the steam and the gases flow in the counter current direction for the first pass and parallel current for the return or second pass. Similarly, with a four-pass superheater, the first and third passes are counter current,

TABLE 7

		Steam Temperatures.		Gas Temperatures.		Calculated Tube Temperatures.				
$\frac{w_g}{a_g}$	$\frac{w_s}{a_s}$	Inlet. °F.	Outlet. °F.	Inlet. °F.	Outlet. °F.	At Steam Inlet. °F.	At Steam Outlet. °F.	Outside Heating Surface, sq. ft.	Length of each Tube, feet.	No. of Tubes.
2·5	25	370	550	2000	700	400	682	130	19·8	21
,,	50	,,	,,	,,	,,	386	622	124	36·0	11
,,	75	,,	,,	,,	,,	381	598	122	55·8	7
,,	100	,,	,,	,,	,,	378	586	121	77·5	5

whilst the second and fourth are parallel current. These conditions modify the calculations somewhat because it becomes necessary to calculate for each pass separately. To do this the best method is first to make a trial calculation for the first pass, assuming a probable value for the steam temperature at the end of the first pass (gas inlet), and then to calculate for the second and other passes in a similar manner. With a few trial calculations temperatures are arrived at giving approximately equal lengths for each pass, and the solution can then be completed. The following Table 8 shows calculated values for 10,000 lb. of steam per hour, entering with dryness fraction ·99 and temperature 370° F., and superheated 550° F. in the first case, and to 620° F. in the second case; using 1-in. bore tubes through which the steam flows.

TABLE 8

	$\frac{w_g}{a_g}$	$\frac{w_s}{a_s}$	Steam Temperatures.		Gas Temperatures.		Calculated Tube Temperatures.		Outside Heating Surface, sq. feet.	Length of Tubes.		No. of Tubes.	No. of Passes per Tube.
			Inlet. °F.	Outlet. °F.	Inlet. °F.	Outlet. °F.	At Steam Inlet. °F.	Highest Value. °F.		Per Pass, feet.	Total, feet,		
1st Case	2·5	25	370	550	2000	700	400	600	152	11·5	23	21	2
	2·0	,,	,,	,,	,,	,,	395	578	178	13·5	27	21	2
2nd Case	2·5	25	370	620	2000	700	400	700	224	8·5	34	21	4
	2·0	,,	,,	,,	,,	,,	395	680	264	10·0	40	21	4

Example VI.—Air Heated by Hot Gases.—In this example hot gases flow through tubes of 1-in. bore and the air flows on the outside of the tubes in the counter current direction as represented in Fig. 37.

The following symbols are used with the given data :—

w_g=gas-flow=30 lb. per sec.

T_{g1}=gas temp. at inlet =800° F.

T_{g2}= ,, ,, ,, outlet=300° F.

θ_T=tube temp. at top or hot end, °F.

θ_B=tube temp. at bottom or cold end, °F.

d_g=inside dia. of tubes (gas side) =1 in.

P_g=mean gas pressure=2116 lb. sq. ft. abs.

T_g=mean gas temp., °F. abs.

ΔP=total resistance to gas-flow =(4 in. water × 5·2)=20·8 lb. sq. ft.

l=length of tubes, ft.

$m_g=\dfrac{\text{area for gas-flow,}}{\text{perimeter}}$ ft.

a_g=area for gas-flow, sq. ft.

v_{g1}=gas velocity at inlet, ft. per sec.

v_{g2}= ,, ,, ,, outlet, ,, ,,

ρ_{g1}=gas density at inlet, lb. per cub. ft.

h_g=rate of heat-flow, B.Th.U. per sq. ft. per sec. per deg. Fahr. diff.

s_g=specific heat of gases=·25, say.

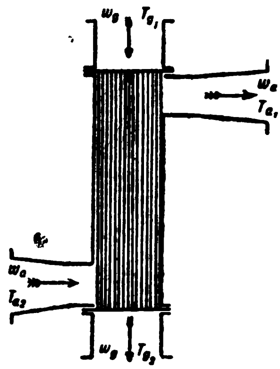


Fig. 37.—Tubular air heater.

Similar symbols are used for the air side of the tubes with the following data :—

T_{a1}=500° F.

T_{a2}=60° F.

P_a=2116 lb. sq. ft. (nearly).

ΔP_a=(3 in. water × 5·2)=15·6 lb. sq. ft.

d_a=1·25 in. (external diameter).

s_a=·24.

It is taken that there are no appreciable losses of heat externally; also, that there is no recovery of the energy of flow at the gas or the air outlets, and that the velocity of flow in the ducts is negligible.

Then, $30 \times \cdot 25 \times (800-300) = w_a \times \cdot 24 \times (500-60)$

$$w_a = \frac{30 \times \cdot 25 \times 500}{\cdot 24 \times 440}$$
$$= 35 \cdot 6 \text{ lb. per sec.}$$

By the same methods as on p. 53, using $f = \cdot 006$, say,

$$\Delta P_g = \cdot 006 \times \frac{53 \cdot 2}{64 \cdot 4} \times \frac{\tau_g}{P_g} \times \frac{l \times 48}{1}\left(\frac{w_g}{a_g}\right)^2 + \frac{1 \cdot 5}{64 \cdot 4}\left(\frac{w_g}{a_g}\right)^2 \times \frac{1}{\rho_{g1}}$$

$$\frac{1}{\rho_{g1}} = 53 \cdot 2 \times \left(\frac{T_{g1}+460}{P_g}\right) = \frac{53 \cdot 2 \times 1260}{2116} = \begin{array}{l}31 \cdot 7 \text{ cu. ft. per lb.}\\ \text{(approximately)}\end{array}$$

$$\tau_g = \frac{800+300}{2} + 460 = 1010° \text{ F. abs.}$$

$$\therefore \quad \Delta P_g = 4 \times 5 \cdot 2 = \left\{\cdot 114 l + \cdot 74\right\}\left(\frac{w_g}{a_g}\right)^2$$

$$\text{or, } l = \frac{182 \cdot 5}{\left(\frac{w_g}{a_g}\right)^2} - 6 \cdot 49 \text{ ft.} \qquad (1)$$

Similarly, for the air side,

$$\Delta P_a = 3 \times 5 \cdot 2 = \cdot 006 \times \frac{53 \cdot 2}{64 \cdot 4} \times \frac{l}{m_a} \times \frac{\tau_a}{P_a} \times \left(\frac{w_a}{a_a}\right)^2 + \frac{1 \cdot 5}{64 \cdot 4}\left(\frac{w_a}{a_a}\right)^2 \frac{1}{\rho_{a2}}$$

$$\tau_a = \frac{60+500}{2} + 460 = 740° \text{F. abs.}$$

$P_a = 2116$ nearly.

$$\frac{1}{\rho_{a2}} = \frac{53 \cdot 2 \times 520}{2116} = 13 \cdot 1 \text{ cub. ft. per lb.}$$

$$\therefore \quad 15 \cdot 6 = \left(\cdot 001735 \frac{l}{m_a} + \cdot 306\right)\left(\frac{w_a}{a_a}\right)^2$$

$$\text{or, } \frac{l}{m_a} = \frac{9000}{\left(\frac{w_a}{a_a}\right)^2} - 176 \text{ ft.} \qquad (2)$$

If $\quad n$ = number of tubes.

$m_g = \frac{a_g}{n \pi d_g}$, and $m_a = \frac{a_a}{n \pi d_a}$ (neglecting the perimeter of the enclosing casing)

$$\therefore \quad n\pi = \frac{a_g}{m_g d_g} = \frac{a_a}{m_a d_a}.$$

$$\text{or,} \frac{a_g}{\frac{1}{48} \times 1} = \frac{a_a}{m_a \times 1\cdot25}$$

$$\text{or,} \quad a_a = 60\, a_g m_a$$

$$\therefore \quad \frac{w_a}{a_a} = \frac{35\cdot6}{60 a_g m_a} = \frac{\cdot593}{m_a a_g} \quad \ldots\ldots\ldots\ldots \quad (3)$$

For a trial calculation take $l = 15$ ft.

Then, from equation 1, $15 = \dfrac{182\cdot5}{\left(\dfrac{w_g}{a_g}\right)^2} - 6\cdot49$

$$\text{or,} \quad \frac{w_g}{a_g} = \sqrt{\frac{182\cdot5}{21\cdot49}} = 2\cdot91$$

$$\text{and,} \quad a_g = \frac{30}{2\cdot91} = 10\cdot3 \text{ sq. ft.}$$

But cross-sectional area per tube $= \dfrac{\pi}{4} \times \dfrac{1}{144}$ sq. ft

$$\therefore \quad n = \frac{10\cdot3 \times 144}{\frac{\pi}{4}} = 1890 \text{ tubes.}$$

From equation 3, $\dfrac{w_a}{a_a} = \dfrac{\cdot593}{m_a \times 10\cdot3} = \dfrac{\cdot0575}{m_a}$,

and inserting this value in equation 2,

$$15 = m_a \left\{ \frac{9000}{\left(\dfrac{\cdot0575}{m_a}\right)^2} - 176 \right\}$$

$$15 = 2{,}730{,}000 m_a^3 - 176 m_a.$$

By trial, $m_a = \cdot0189$ ft.

$$\therefore \quad \frac{w_a}{a_a} = \frac{\cdot0575}{\cdot0189} = 3\cdot04 \equiv 3, \text{ say.}$$

$$\text{and,} \quad a_a = \frac{w_a}{3} = \frac{35\cdot6}{3} = 11\cdot9 \text{ sq. ft.}$$

Neglecting conduction of heat along the length of the tubes and the small drop of temperature through the tube.

$$h_g(\mathrm{T}_{g1} - \theta_\mathrm{T})\pi d_g = h_a(\theta_\mathrm{T} - \mathrm{T}_{a1})\pi d_a$$

$$\text{or,} \quad \theta_\mathrm{T} = \frac{h_g \mathrm{T}_{g1} d_g + h_a \mathrm{T}_{a1} d_a}{h_g d_g + h_a d_a}$$

and, $\theta_B = \dfrac{h_g T_{g2} d_g + h_a T_{a2} d_a}{h_g d_g + h_a d_a}$

If $h_g = \cdot0035$, and $h_a = \cdot0035$

Then, $\theta_T = \dfrac{\cdot0035 \times 800 \times 1 + \cdot0035 \times 500 \times 1\cdot25}{\cdot0035 \times 1 + \cdot0035 \times 1\cdot25}$

$= \dfrac{2\cdot8 + 2\cdot18}{\cdot00788} = 632° \text{ F.}$

$\theta_B = \dfrac{\cdot0035 \times 300 \times 1 + \cdot0035 \times 60 \times 1\cdot25}{\cdot0035 \times 1 + \cdot0035 \times 1\cdot25}$

$= \dfrac{1\cdot05 + \cdot262}{\cdot0035 \times 2\cdot25} = 167° \text{ F.}$

Calculation gives the probable mean difference between gas and tube 150° F.,

and, Total tube surface on gas side $= \pi \times \frac{1}{12} \times 15 \times 1890$
$= 7410$ sq. ft.

Then,

$h_g \times 150 \times 7410 =$ heat supplied to tubes $= 30 \times \cdot25 \times (800 - 300)$

or, $h_g = \dfrac{30 \times \cdot25 \times 500}{150 \times 7410} = \cdot00337$

This is sufficiently near the value for $h_g = \cdot0035$ adopted originally to justify the adoption of 1890 tubes, 1-in. bore, of length about 15 ft.

Heat transmission per sq. ft. per hour

$= \dfrac{30 \times \cdot25 \times 500 \times 3600}{7410} = 1820$ B.Th.U.

Example VII.—**Heat Transmission from Gases passing through a Clean Brickwork Flue.**—The arrangement relating

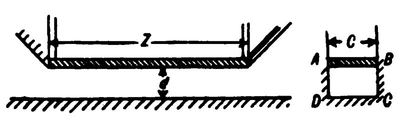


Fig. 38.—Boiler plate subject to radiation from brick walls.

to this example is represented in Fig. 38, one side of which is a plate forming part of a boiler shell. In these calculations it will be taken that there is no appreciable conduction of heat

through the brickwork, and that the brickwork cavity is shallow.

Let T_1 = gas temp. at inlet $= 1500°$ F.

T_2 = „ „ „ outlet $= 600°$ F.

t_s = steam or water temp. at boiler $= 360°$ F.

T_{b1} = brickwork temp. at inlet end, °F.

T_{b2} = „ „ „ outlet „ °F.

d = depth of flue, ft.

c = width of flue, ft.

l = length of flue, ft.

$$\frac{w_1}{a_1} = \frac{\text{gas-flow, lb. per sec.}}{\text{area of flue section, sq. ft.}} = 1$$

h = rate of heat transmission from gas to water
= „ „ „ „ „ „ to brickwork
$= \cdot 0015$ B.Th.U. per sq. ft. per sec. per deg. Fahr. difference

*e = coefficient of radiation for clean brickwork
$= \cdot 5$, say.

In the length of flue δx feet the heat lost by radiation from the brickwork to the plate per second is,

$$e \times \frac{1600}{3600}\left\{\left(\frac{T_b+460}{1000}\right)^4 - \left(\frac{t_s+460}{1000}\right)^4\right\} c\,\delta x. \quad \text{B.Th.U.}$$

taking it that the gases do not absorb much of the radiant energy.

$$\text{Then, } h(T-T_b)c\,\delta x = e \times \frac{1600}{3600}\left\{\left(\frac{T_b+460}{1000}\right)^4 - \left(\frac{t_s+460}{1000}\right)^4\right\} c\,\delta x$$

Total heat received by the plate per second in length δx.

$$= h(T-t_s)c\,\delta x + e \times \frac{1600}{3600}\left\{\left(\frac{T_b+460}{1000}\right)^4 - \left(\frac{t_s+460}{1000}\right)^4\right\} c\,\delta x$$

$$= hc\,\delta x\{(T-t_s)+(T-T_b)\}$$

Thus at inlet,

$$\cdot 0015(1500-T_b) = \cdot 5 \times \frac{1600}{3600}\left\{\left(\frac{T_b+460}{1000}\right)^4 - \left(\frac{360+460}{1000}\right)^4\right\}$$

By trial it is found that $T_b = 960°$ F.

and therefore, Heat given in length δx

$$= \cdot 0015c\,\delta x\{(1500-360)+(1500-960)\}$$
$$= \cdot 0015c\,\delta x\ (1140+540)$$

* This value is not known except very approximately.

Thus, the effect of radiation has been to increase the effective value of h by $\frac{540 \times 100}{1140} = 47\%$ approximately.

Similarly, at the outlet h is apparently increased by about 29%.

It may be taken for the present purpose that the average increase of h under these conditions is, say, 35%.

$$\therefore \; h' = h \times 1{\cdot}35 = {\cdot}0015 \times 1{\cdot}35$$
$$= {\cdot}002$$

Heat transmitted per sec. $= w_1 \times {\cdot}25(1500-600)$
$= w_1 \times 225$ B.Th.U.

$$\text{Mean temperature difference} = \frac{1500-600}{\log_e \dfrac{1500-360}{600-360}} = 578^\circ \text{F. nearly.}$$

Then, $\qquad A \times {\cdot}002 \times 578 = w_1 \times 225$

or, area of heating surface, $A = \frac{w_1 \times 225}{{\cdot}002 \times 578} = 195 w_1$ sq. ft.

and, length of flue, $\qquad l = \frac{A}{c} = \frac{A \times d}{a_1}$

$$= 195 d \frac{w_1}{a_1} \text{ ft.}$$

In this case, $\frac{w_1}{a_1} = 1$, and thus, $l = 195d$.

If $d = \frac{1}{3}$ ft., for instance, $\qquad l = \frac{195}{3} = 65$ ft.

Example VIII.—In the previous example (VII) the brickwork was clean, but if covered with soot the walls would probably radiate nearly as a black body, in which case $e=1$ nearly.

Thus, at inlet

$${\cdot}0015\,(1500 - T_b) = \frac{1600}{3600}\left\{\left(\frac{T_b+460}{1000}\right)^4 - \left(\frac{360+460}{1000}\right)^4\right\}$$

By trial, $T_b = 827^\circ$ F.

and, ${\cdot}0015\{(1500-360)+(1500-827)\}c\,\delta x$
$=$ heat given in length δx.
$= {\cdot}0015\,(1140+673)\,c\,\delta x$.

Therefore, the effect of radiation at inlet is to increase the value of $h = \cdot 0015$ to $h' = \cdot 0015 \times \left(1 + \frac{673}{1140}\right)$

$$= \cdot 0015 \times 1 \cdot 59$$
$$= \cdot 00238.$$

At outlet,

$$\cdot 0015\,(600 - T_b) = \frac{1600}{3600}\left\{\left(\frac{T_b + 460}{1000}\right)^4 - \left(\frac{360 + 460}{1000}\right)^4\right\}$$

By trial, $T_b = 490°$ F.

and, $\cdot 0015\{(600 - 360) + (600 - 490)\}c\,\delta x$

$$= \cdot 0015\,(240 + 110)\,c\,\delta x,$$

an increase of $h = \cdot 0015$ to $h' = \cdot 0015\left(1 + \frac{110}{240}\right)$

$$= \cdot 0015 \times 1 \cdot 46.$$

Probable mean value of $h' = \cdot 0015 \times 1 \cdot 5 = \cdot 00225$, say.

Then, $A \times \cdot 00225 \times 578 = w_1 \times 225$, as on p. 97.

$$\text{or,} \quad A = \frac{w_1 \times 225}{\cdot 00225 \times 578} = 173 w_1$$

$$\text{and} \quad l = \frac{A}{c} = 173 d \frac{w_1}{a_1} \text{ ft.}$$

With $\frac{w_1}{a_1} = 1$, $\qquad l = 173 d$ ft.

and, if $d = \frac{1}{3}$ ft., then $\qquad l = \frac{173}{3} = 58$ ft.

It follows also that to make effective use of the radiations from a brickwork wall, the cavity should be shallow and wide.

Some Empirical Formulæ Relating to Heat Transmission.—From the results of experiments on the evaporative power of boilers Professor Rankine* originally proposed the following formula as a rough approximation for the heat transmission in boilers.

$$H = \frac{(T - t)^2}{b}$$

where H is in B.Th.U. per sq. ft. per hour, T is the gas temperature and t the water temperature in deg. Fahr., and b was taken to lie between 160 and 200.

* *The Steam Engine.*

Though the above expression may represent the results obtained within the limited range of application on which it was based, it is evident from all the experiments discussed that such an expression cannot be of general application, and fails completely when tested against actual results obtained at widely different rates of gas-flow.

Although there is evidence to show that a few engineers early recognised that the velocity of the gas had some influence on the rate of heat transmission, the following quotation indicates that most engineers were completely ignorant of the laws of heat transmission until within comparatively recent years. In his article on "Heat Transmission in Boilers,"* J. G. Hudson said: "The element of speed has not, so far as the author is aware, ever been hitherto taken into account, as possibly affecting the transmission. In the few cases where reference has been made to it, a high speed has been considered objectionable, as reducing the time available for the gases to part with their heat. . . ."

From his analysis of a number of experiments on boilers Mr. Hudson proposed the following formula :—

$$h = \frac{T+t+922}{2} \times \frac{\sqrt{v}}{B}$$

where h = B.Th.U. per sq. ft. per hour per deg. Fahr. difference.

T = gas temperature, °F.

t = water temperature, °F.

v = velocity of gas, ft. per sec.

$B = 1250$

It is hardly necessary to show that the above formula is not of general application, but probably it can be made to represent boiler conditions over a much wider range than the Rankine formula.

The author has not thought it worth while to discuss the above formulæ any further because of their empirical nature and the limited scope of their application.

* *The Engineer*, Dec. 12th, 1890.

CHAPTER II

CONDENSATION OF STEAM AND RATE OF HEAT TRANSMISSION IN SURFACE CONDENSERS AND COOLERS

Condensation of Steam.—There are two principal types of steam condensers in common use: (1) Those condensers in which the cooling substance, usually water, comes into direct contact with the steam, as in the various arrangements of jet condensers, and (2) Those in which the steam and the cooling substance, usually circulating water, are separated by metal surfaces, and are generally termed "surface condensers." In this latter type the circulating or cooling water is usually passed through brass tubes and the steam condensed on the outside surface of these tubes; but in air-cooled condensers and in evaporative condensers the steam is commonly condensed inside the tubes, and the cooling substance, in the first-named case atmospheric air, and in the second, water trickling over the surfaces and evaporating, is in contact with the outer surface.

In what follows, the discussion is strictly limited to the consideration of the transmission of heat in surface condensers, since it is in these condensers only that there is any serious problem associated with the transmission of heat from the condensing steam to the cooling or refrigerating substance. The most important condensers of this class are those used for the condensation of the exhaust steam from steam engines and turbines, wherein the steam is condensed at a comparatively low temperature with the object of reducing the back pressure on the engine or turbine to the lowest values obtainable with all-round economy. These condensers require "air pumps" to extract the water of condensation and the air which finds its way into the condenser, and to discharge these

against the pressure of the atmosphere. In this manner the back pressure at the engine or turbine can be reduced to a value much below that of the atmosphere, resulting in a lower steam consumption for a given power developed than could be attained if the exhaust steam were discharged to the atmosphere.

The problems associated with the condensation of steam are generally complicated by the air which enters the condenser with the steam. No matter how carefully the various joints in an exhaust pipe may be made, and however well designed and constructed the low-pressure glands may be, it is practically impossible to secure absolutely air-tight conditions for any length of time. Also, a small amount of air originally in solution in the boiler feed finds its way into the condenser along with the steam. It will be shown later that the air in the condenser tends to increase the resistance to heat-flow on the steam side of the tubes, or, in other words, it tends to retard the transmission of heat when compared with air-free steam, and also, according to the laws of mixtures, the presence of air reduces the steam temperature for any given total pressure in the condenser.

Dalton's Law of Mixtures.*—According to this law the pressure of a gaseous mixture on the walls of the containing vessel is equal to the sum of the pressures which the constituents would exert if each occupied the vessel separately. That is, if p_1, p_2, p_3, etc., are the absolute pressures which would be exerted by the various gases if introduced separately into the containing vessel, then the total pressure due to the mixture would be,

$$p_t = p_1 + p_2 + p_3 + \text{etc.}$$

For high pressures, outside the range of ordinary practice, this relation is not quite true.

In the case of vapours this law is obeyed approximately within certain limits which it is hardly necessary to specify here† as these limits are outside the range of practice with which we are at present concerned.

Therefore in a mixture of air and water vapour the total

* Dalton, *Memoirs of Manchester Phil. Soc.*, Vol. V, 1802, p. 543.

† Refer to Preston's *Theory of Heat* for these conditions.

pressure p_t is the sum of the "partial" pressures due to the air p_a and the vapour p_s, that is,

$$p_t = p_a + p_s.$$

When a space already occupied by air or any other gas is saturated with water vapour, the partial pressure exerted by the vapour is very nearly equal to the pressure which the vapour would exert in vacuo at the same temperature. It has also been proved by experiment that the weight of vapour per unit volume at any particular temperature is the same whether in vacuo or mixed with a gas.

Thus, when steam enters a condenser it usually carries with it some air which has leaked into the system, and as the mixture of steam and air flows through the condenser it is obvious that the ratio of the weight of air to the weight of steam increases gradually from the condenser inlet to the air pump suction. Provided the temperature and pressure of the mixture are known at any point accurately it is possible to estimate the ratio of air to steam at that point if the assumption is made that the air is just saturated with vapour. For example, if the total absolute pressure is 3 in. of mercury and the temperature 100° F. the calculation is as follows: At 100° F. the saturation steam pressure is 1·93 in. of mercury, and thus the air pressure is $3-1{\cdot}93=1{\cdot}07$ in. Since $\frac{\mathrm{PV}}{\tau}=53{\cdot}2$ for 1 lb. of air, where P=air pressure, lb. per sq. ft., V=cub. ft. per lb., and τ=absolute temperature, deg. Fahr.,

$$\text{Then, } \frac{1{\cdot}07 \times {\cdot}491 \times 144 \times \mathrm{V}}{(460+100)} = 53{\cdot}2$$

$$\text{or, } \quad \mathrm{V} = 394 \text{ cub. ft.}$$

The density of dry saturated steam at 100° F. is ·00285 lb. per cub. ft. Thus, the weight of steam associated with 1 lb. of air saturated at 100° F. and having the total pressure 3 in. of mercury is $394 \times {\cdot}00285 = 1{\cdot}12$ lb., and the ratio of air to steam is $\frac{1}{1{\cdot}12} = {\cdot}89$ by weight.

In an actual condenser it is hardly likely that the mixing of air and vapour is quite complete, and there may be local differences in the conditions as the air and steam pass through the condenser. On the assumption, however, that the air is always saturated the above method of calculation has been

used by Mr. D. B. Morison* to obtain the graph shown in Fig. 39. The ordinates represent the volumes of 1 lb. of air saturated with water vapour at various temperatures

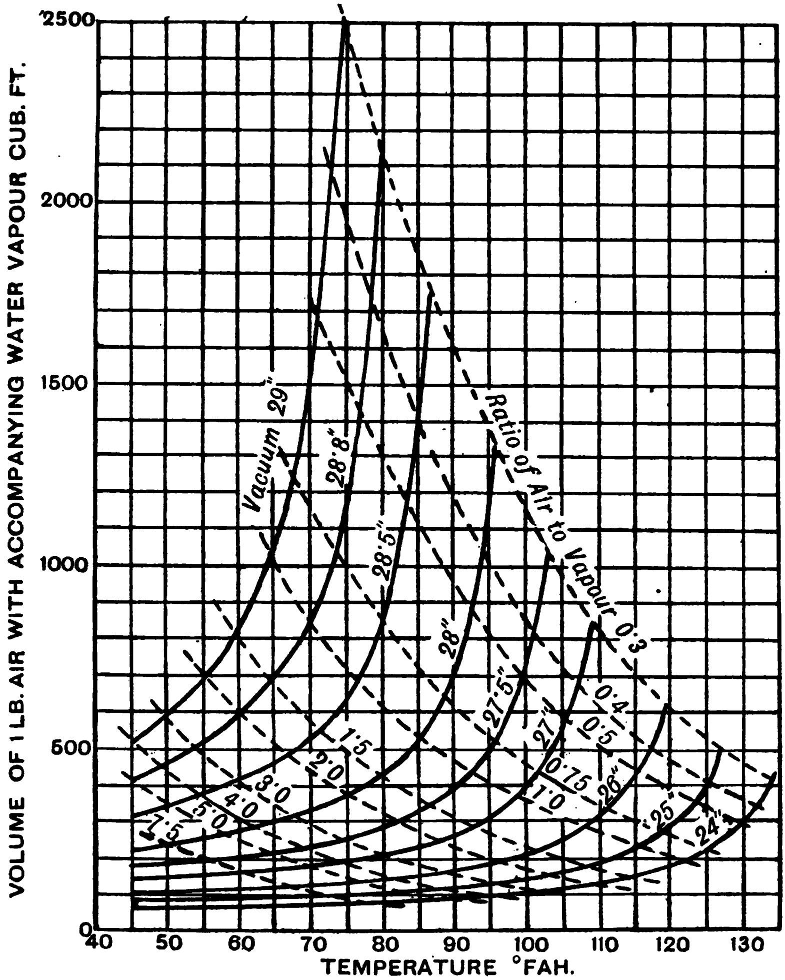


FIG. 39.—Volumes of mixture of aii and vapour per pound of air.

and vacua when the barometric pressure is 30 in. of mercury. The full lines represent a series of constant vacua, while

* "The Influence of Air on Vacuum in Surface Condensers," *Inst. of Naval Arch.*, 1908.

the dotted lines refer to constant ratios of air-weight to the weight of accompanying water vapour. Following any constant vacuum line it is seen how an increase in the ratio of air to steam as the steam gradually condenses causes a corresponding reduction in the temperature of the mixture. In other words, cooling and " de-vaporising " the air at constant vacuum rapidly reduces the volume per pound of air

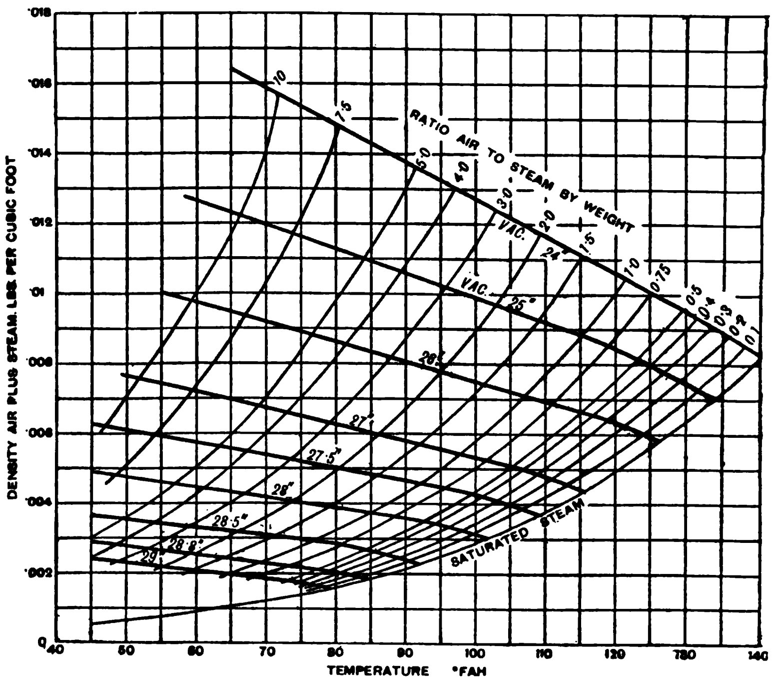


FIG. 40.—Density of mixture of air and vapour.

dealt with and therefore reduces the corresponding displacement of the air pump proportionally.

Fig. 40 has been constructed to illustrate how the density of the mixture of air and saturated vapour varies with the temperature at different constant vacua, indicated by the heavy lines. The light lines refer to constant ratios of air to steam by weight. Following any constant vacuum line it is seen that as the ratio of air to steam increases the density of the saturated mixture increases, which suggests that the natural

position for the air-pump suction is at the bottom of the condenser and the steam inlet at the top. It is probable, however, that under actual conditions in practice, the small

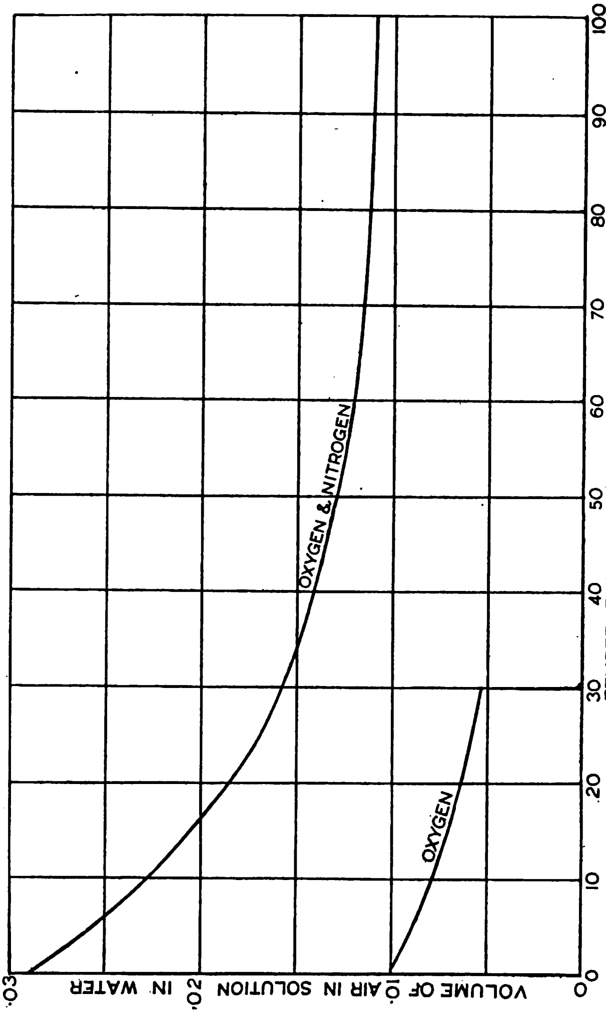


Fig. 41.—Air in solution in water at different temperatures.

increase in vacuum which usually occurs from the steam inlet to the air pump suction and the rush of the current of steam and air through the condenser, would render it almost as easy

to extract the air from the top if it happened to be convenient to have the steam inlet at the bottom. With such an arrangement, however, it might be found more difficult to drain the water of condensation from the tubes.

Air in Solution in Water.—The solution of a gas in a liquid may be taken to be a mixture of two liquids, the dissolving liquid itself and the dissolved gas. Henry's* law states that the volume of a gas absorbed in a saturated liquid at any particular temperature is independent of the pressure of the gas. Dalton introduced the corollary that in a mixture of gases the volume of any of the constituents of the mixture absorbed in a saturated liquid is independent of the partial pressure. The amount of a gas absorbed depends, of course, on its solubility in the liquid, which decreases with an increase of temperature of the liquid.

According to Bunsen the gases absorbed from the air (neglecting carbon dioxide) by pure water consists of a mechanical mixture of oxygen and nitrogen in the constant ratio of 34·91 oxygen to 65·07 nitrogen.

According to Winkler† the volume of air required to saturate one volume of water is given in Fig. 41, on the base of temperatures in degrees Cent. It would be noticed that at 100° C., water, when saturated with air, contains over one-third the volume at 0° C. On boiling, of course, the whole of the air would come out of solution from the water. In the same diagram up to 30° C. is shown the volume of oxygen included in the air in solution, varying from 34·9% of the volume of air at 0° C. to 33·6% at 30° C.

In practice, however, the water might not be saturated, or again it might have a certain amount of air entrained in addition to that held in solution. The values in Table 9 are given by Mr. J. A. Smith‡ from the results of experiments made by him. Unless otherwise mentioned the measurements were made at, or reduced to, 54° F. and 29·9 in. of mercury.

* This law does not hold, however, if the gas is very soluble, as for instance ammonia gas in water, but for gases like oxygen and nitrogen in water it is true.

† Landolt and Börnstein's Tables.

‡ "Air in Relation to Boiler Feeds," *Inst. Victorian Engineers*, 1904; or *Engineering*, Oct. 7th, 1904.

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
11. Melbourne city service supply, enclosed in glass jar with 2 oz. fresh green grass per lb. water, placed in open for twenty-four hours at 54° F. . . .	·0223	7·0	17·1	75·9

It will be noticed that fresh water at ordinary atmospheric temperature is likely to contain slightly over 2% by volume of dissolved gases measured at atmospheric pressure and temperature. A certain amount of corrosion is probably caused by the liberation of free oxygen when in contact with iron, as is indicated by the loss of oxygen in lines 9 and 10 in contact with clean cast-iron borings. The presence of carbon-dioxide in water also probably promotes corrosion, and line 11 suggests that in river water the presence of carbon-dioxide may be largely due to vegetation in the water.

Mr. Smith also measured the volume of gas driven out of solution by heating water to various temperatures starting with water at 54° F. and barometric pressure 29·9 in. of mercury, and heating it to 212° F. in about 150 minutes. The first evolution of gas occurred at about 120° F. It is seen from the values given in Table 10 that the evolution of gas is quite small up to about 170° F.

TABLE 10

Temperature, °F.	140	175	190	200	210	212
Volume of gas evolved up to this temperature per unit volume of water (measured at 54° F. and 29·9 in. mercury).	·0001	·001	·002	·006	·0165	·022

In his paper on "Air in Surface Condensation," Mr. G. A. Orrok* gives some results of his investigations in the Waterside No. 2 station of the New York Edison Company. He determined the amount of air in solution and mechanically entrained in the feed water by absorbing the oxygen chemically and estimating the free iodine liberated by titration with a solution of sodium theosulphate of known strength, and from this the amount of oxygen in solution was calculated from the known reactions. The process to the liberation of iodine is expressed by the following chemical equations :—

$$2\ Mn(OH)_2+O_2=2H_2MnO_3$$
$$2H_2MnO_3+8HCl=2MnCl_2+6H_2O+2Cl_2$$
$$4KI+2Cl_2=4KCl+2I_2.$$

* *Journal Amer. Soc. Mech. Engs.*, Nov., 1912.

The amount of nitrogen was estimated from the usual proportions of oxygen to nitrogen absorbed in water. The following are some of the results at atmospheric pressure :—

Mechanically entrained in feed water at 187° F.	·00015 vol. of water
In solution in feed water at 187° F. . .	·00916 ,,
∴ Total air in feed water from open heaters, 187° F. . . .	·00931 ,,
In solution in Croton water supply (make-up feed), 52° F.	·04325 ,,
In solution in hot-well water, 80° F. .	·00269 ,,

According to Mr. D. B. Morison* the water of condensation discharged by the air-pump from a surface condenser, when used over and over again in the same plant, settles down to a permanent charge of about 2 volumes of air at atmospheric pressure per 100 volumes of water, most of which comes out of solution in the boiler and is reabsorbed in the condenser and as the water passes through the air-pumps. He refers, probably, to ordinary marine practice, where the water is commonly heated in closed feed heaters before entering the boiler. As has been shown, heating the feed water in the open to near 212° F. would liberate most of this air, and some exhaust feed heaters have arrangements for this purpose.

Air Leakage in Condensers.—The air in solution in the feed water is only a small proportion of the air which has to be extracted by air-pumps in ordinary surface condenser practice. For example, taking the air in solution in the feed water to be 2% by volume at atmospheric pressure (14·7 lb. sq. in.) and temperature (60° F.) this volume is

$$\frac{\cdot 02}{62\cdot 4} = \cdot 00032 \text{ cub. ft. per lb. of water.}$$

If the partial pressure of this air in the condenser is ·744 lb. sq. in. and temperature 100° F. then the volume of the air becomes

$$\cdot 00032 \times \frac{14\cdot 7}{\cdot 744} \times \frac{(460+100)}{(460+\ 60)} = \cdot 00684 \text{ cub. ft.}$$

per lb. of water of condensation. But according to Professor

* The "Influence of Air on Vacuum in Surface Condensers," *Inst. Naval Arch.*, 1908.

Weighton's* experiments, for example, an air-pump suction stroke capacity of nearly 0·7 cub. ft. per lb. of steam condensed is required with a reciprocating engine, even when precautions are taken to prevent air leakage into the condenser. Even allowing for the low volumetric efficiency of ordinary air-pumps a comparison of the above values shows what a small proportion of the required pump displacement is due to the air in solution in the feed water.

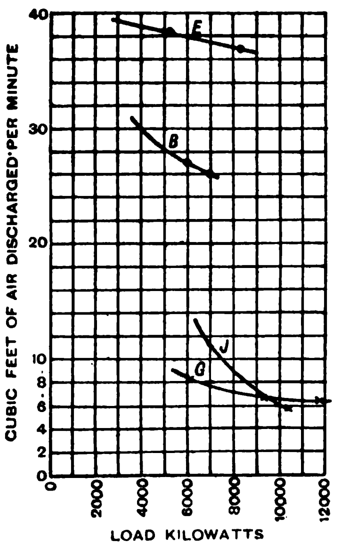


FIG. 42.—Air leakage into condensers at various loads on turbines.

In the paper referred to on p. 108 Mr. Orrok gave some results of experiments he had made on the quantity of air discharged from surface condensers connected to steam turbines. The discharge pipe from the dry air-pumps was connected up through a valve to a small bell gasometer located near the air pumps. After the air-pump had been running for a sufficient length of time to make sure of constant conditions the valve was opened and the air-pump then discharged into the bell, allowing it to rise, which would take from 1 to 20 minutes to fill, according to circumstances. The air in the bell was brought to atmospheric pressure and temperature before the final readings were taken.

Four sets of tests were run to determine the effect of the power load on air leakage, two on turbines which were in bad condition as regards air leakage, marked E and B in Fig. 42, and two on comparatively tight machines, marked J and G. It will be noted that in every case the air leakage was less at the higher loads than at light loads. The tube surfaces in these condensers were respectively, E, 18,000 sq. ft.; B, 21,000 sq. ft.; G, 18,000 sq. ft.

* "The Efficiency of Surface Condensers," *Inst. Naval Arch.*, 1906.

To ascertain the effect of known quantities of air on the vacuum, circular sharp-edged orifices of various sizes were placed at the top of the condenser so that the air introduced might mix with the incoming steam. These results are shown in Fig. 43, the load on the turbines being indicated on the diagram. Condenser H had a tube surface of 18,000 sq. ft.

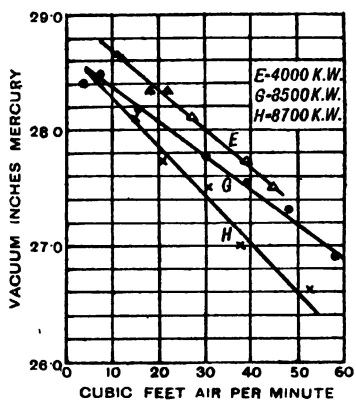


FIG. 43.—Vacua obtain at various rates of air leakage.

The conclusions arrived at were that the air discharged by the dry air-pump at atmospheric pressure and temperature from units between 5000 K.W. and 20,000 K.W. in size, varies from 1 cub. ft. per min. at atmospheric pressure and temperature when in the best condition, to 15 or 20 cub. ft. where ordinary leakage is present, and to 30 or perhaps 40 or 50 cub. ft. where air leakage is bad. Most of this leakage comes into the condenser through minute leaks in the cast-iron shell, pipe joints, and expansion joints, which it is practically impossible to detect except by filling the condenser with warm water and subjecting it to a pressure of a few pounds per sq. in. above the atmosphere.

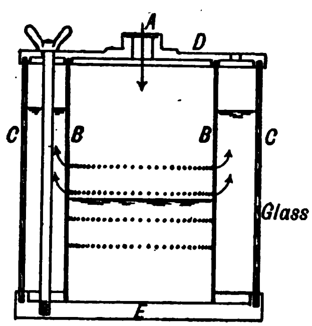


FIG. 44.—One form of Weighton's air indicator.

Usually it is taken that the leakage of air in steam turbines with short exhaust connections is about 2 to 3 lbs. of air per 10,000 lbs. of steam at normal full load, and in reciprocating engines two or three times this amount. With multiple effect evaporators the air to be dealt with may amount to 3 or 4 lbs. per 1000 lbs. of vapour entering the condenser.

A useful type of air leakage indicator, introduced by Professor R. L. Weighton, is illustrated in section in Fig. 44. It

consists of a vessel or bell B, either fixed or floating inside the glass casing C, and the air-pumps discharge the air into the bell. If the bell is fixed the air enters at the top as shown in Fig. 44, but if made to float the air is arranged to enter from below. The bell B is perforated by a large number of small holes through which the air leaks to the atmosphere, the rate of air-flow being indicated by the number of holes showing leakage or by the difference of level between the water inside and outside the bell B. If the bell is floating in the water the position of the bell, indicated on a scale, is a measure of the number of holes through which air is flowing.

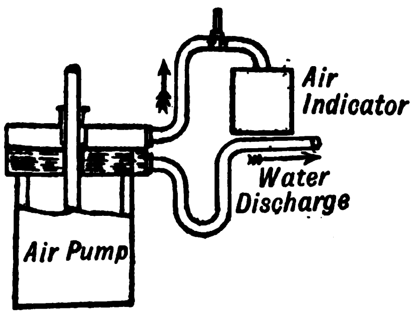


FIG. 45.—One arrangement of connections of air indicator to air pump.

The general arrangement of such an air indicator is illustrated in Fig. 45, where the wet air-pumps discharge the water of condensation at the same or at a slightly lower level than that at the head of the air-pumps. If it was necessary to discharge the water at a higher level then a separator could be introduced at or near to the highest point of the discharge pipe to separate the water and the air discharged. Whilst this indicator is used to indicate directly any increased or abnormal leakage of air in the condenser it may also be calibrated to indicate the actual leakage or volume of air discharged. One approximate method with reciprocating air-pumps is to indicate the air-pumps and to calculate from the diagram the amount of air discharged, comparing this with the reading on the air indicator scale.

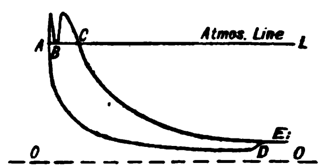


FIG. 46.—Diagram from Edwards air pump

Considering the diagram shown in Fig. 46, taken from an Edwards air-pump, AL, the length of the diagram, represents the stroke volume of the pump to some scale. EC is the compression line and BD the re-expansion line. Then the length

CB between the points where these lines cut the atmospheric line represents the volume discharged at atmospheric pressure, assuming the leakage at the air-pump piston to be negligible and the indicator diagram to be a true record of the pressures and volumes in the cylinder.

Let A = area of pump barrel, sq. ft.
L = length of stroke, ft.
l = length of indicator diagram, AL in Fig. 46.
c = length CB on diagram.
N = number of suction strokes per minute.
t = temperature of discharge, °F.
W = weight of water of condensation discharged per min., lb.

Then, Volume* equivalent to CB $= \frac{c}{l} \times A \times L$, cub. ft.

and, Volume* per minute $= \frac{c}{l} \times A \times L \times N$ cub. ft.

With wet air pumps the approximate volume of the water of condensation discharged is $\frac{W}{62}$ cub. ft. per min. and this could be deducted from the previous expression to determine the volume of air.

It may be taken that the temperature of the air at C and B is the same as the discharge temperature, which, however, may not be quite true, but an error of a few degrees in temperature at these points does not greatly affect the result of the calculation. Particularly when the air discharge is small, and therefore the distance CB small, there is sometimes difficulty in estimating the exact length of CB on account of erratic changes in the compression line EC from cycle to cycle. The best that can be done is to take a fairly large number of diagrams at equal intervals of time and estimate the average length of CB.

The weight of air discharged can also be estimated from the diagram, taking the air to be at atmospheric pressure and saturated with water vapour at the temperature of discharge.

* A certain amount of air bubbles will be entrained in the water filling the clearance space, and on re-expansion along BD will increase in volume. This has been neglected in the following calculations.

If, for example, this temperature is 100° F, the vapour pressure is ·946 lb. per sq. in. If atmospheric pressure is 14·7 lb. per sq. in. then the partial pressure of the air is 14·7—·946=13·75 lb. per sq. in. nearly. The volume of 1 lb. air at this temperature and pressure is $V=\frac{53{\cdot}2\times(460+100)}{13{\cdot}75\times144}=15{\cdot}07$ cub. ft. Thus the weight of air discharged per minute from the wet air-pump would be $\left(\frac{cALN}{l}-\frac{W}{62}\right)\times\frac{1}{15{\cdot}07}$ lb., and, neglecting the water vapour carried away by the air, the approximate ratio of air to steam by weight at condenser inlet is $\left(\frac{cALN}{l}-\frac{W}{62}\right)\frac{1}{15{\cdot}07W}$.

The weight of vapour carried away by the air per minute would be $\left(\frac{cALN}{l}-\frac{W}{62}\right)\times\frac{1}{351}$, lb., where the volume of 1 lb. dry saturated steam at 100° F. is taken to be 351 cub. ft. Compared with the magnitude of W the above weight of water vapour is very small, for taking $ALN=\cdot7\times W$ as an extreme value at full load (the ratio referred to on p. 110), and assuming the large value for $\frac{c}{l}=\cdot2$, say, then, $\left(\frac{cALN}{l}-\frac{W}{62}\right)\frac{1}{351}=$ ·00035W, or, ·035% of W, which is a negligible quantity.

A more accurate method of estimating the volume discharged by the air pumps than by using the indicator diagrams would be to arrange a small gasometer on the air discharge pipe, in much the same manner as in Orrok's experiments described on p. 110. This method could also be used to calibrate the Weighton air indicator, which could then be used to measure the actual volume discharged.

The above discussion regarding the amount of vapour discharged by the air pumps refers to wet air-pumps and also to dry air-pumps supplied with water for sealing and cooling purposes. When a steam or water ejector is used to extract and discharge the air, the above calculation needs to be modified as follows : For example, suppose the air and vapour temperature at the ejector suction is 96° F., and that the air is saturated with water vapour. Then, if the capacity of the ejector is

C cubic feet per minute at condenser pressure, and W is the weight of steam condensed per minute in the condenser, $\frac{C}{393}$ lb. of vapour is extracted per min. by the ejector, where the volume of 1 lb. steam at 96° F. is 393 cub. ft. or, $\frac{C}{W \times 393}$ lb. of vapour per lb. of steam condensed.

If $\frac{C}{W} = 1$, say, then,

$\frac{C}{393W} = \cdot 00255$ lb., or $\cdot 255\%$ of the water of condensation.

With condensers designed for high vacua it is usual to extract the mixture of air and vapour and the water of condensation by separate pumps, and to provide a special set of tubes for cooling the air and vapour (that is, "de-vaporising" as it is sometimes called) below the temperature of the water of condensation. This has a considerable influence on the necessary volumetric capacity of the dry air-pumps or the ejectors, as was shown to be the case in Fig. 39. For example, if the mixture of air and vapour were cooled from 96° F. to 70° F. with the total pressure 2 in. of mercury, the vapour pressure at 70° F. would be ·739 in. and the partial air pressure $2 - \cdot 739 = 1 \cdot 261$ in. The volume of 1 lb. air under these conditions is,

$$V = \frac{53 \cdot 2 \times (460 + 70)}{1 \cdot 261 \times \cdot 491 \times 144} = 316 \text{ cub. ft.}$$

But at 96° F. and the same total pressure 2 in. of mercury the partial air pressure is only ·294 in., and the volume per lb. air is then 1425 cub. ft. The cooling from 96° F. to 70° F. therefore results in a reduction of the volume from 1425 to 316 cub. ft. per lb. of air dealt with.

Thus, if C′ cub. ft. per min. is the necessary capacity of the ejector dealing with the mixture at 70° F., and C that at 96° F.,

$$\frac{C'}{C} = \frac{316}{1425} = \cdot 222.$$

Again, the volume of 1 lb. steam at 70° F. is 871 cub. ft., and the weight of vapour extracted per min. becomes $\frac{C'}{871}$ lb.

per min. With $\frac{C'}{W} = \cdot 222$, then this becomes $\frac{\cdot 222}{871} = \cdot 000255$ lb. per lb. of water of condensation, or ·0255%, or only $\frac{1}{10}$ of the previous amount. It is therefore evident that under ordinary conditions of operation the weight of vapour extracted by the air-pumps is negligible when compared with the water of condensation.

No account has been taken, however, of possible entrainment of water particles by the mixture of air and vapour passing to the air pumps.

M. Leblanc* has estimated the weight of entrained particles of water by calculations from the adiabatic compression of a mixture of air, vapour, and water at different suction temperatures T_1 and delivery temperatures T_2. His estimates, converted to British units, are given in Table 11, and the writer has added the values expressing the amount of the entrained water as a percentage of the water of condensation, taking it that the dry air pump effective displacement is about ·25 cub. ft. per lb. of water of condensation, which represented the conditions in a fairly air-tight steam turbine system.

TABLE 11

α=estimated weight of water particles per 100 cub. ft. of air at air-pump suction.
β=estimated percentage of water particles to water of condensation, assuming an effective air-pump displacement of ·25 cub. ft. per lb. of steam condensed.
T_1=suction temperature, ° F.
T_2=discharge temperature, ° F.

T_2 °F.	T_1							
	50° F.		59° F.		68° F.		77° F.	
	α lbs.	β per cent.	α lbs.	β per cent.	α lbs.	β per cent.	α lbs.	β per cent.
122	·719	·18	1·52	·38	2·26	·56	3·58	·89
140	·675	·17	1·02	·25	1·55	·39	2·24	·56
158	·409	·10	·625	·15	·89	·22	1·29	·32
176	·084	·02	·161	·04	·22	·046	0·36	·09

* "A Note on Condensation," *Engineering*, Aug. 28, 1908.

The discharge temperature from a single stage dry air-pump is commonly about 140° F. and the above calculations would suggest that the entrained weight of water particles per pound of water of condensation varies, roughly, from about ·0017 lb. to ·0056 lb.

A common method of testing the air-withdrawing capacity of an air-pump or ejector is to introduce a suitable orifice or nozzle at a convenient point in the condenser, and either have the condenser working under the ordinary conditions of operation, or else completely blanked off except for the orifice. In the experiments mentioned on p. 111 sharp-edged orifices had been used for this purpose, but for such orifices the coefficient of discharge is not known with any degree of certainty when the difference of pressure on the two sides of the orifice is comparatively large. It is much better to use a convergent nozzle with rounded entrance ; the coefficient of discharge for such a nozzle is about ·98. The quantity of air passing through the nozzle when the absolute pressure on the exit side is less than ·528 of that at the entrance, which is the usual condition in condensers, can be estimated in the following manner :—

Let w=weight of air per second, lb.

a=area at throat (smallest section), sq. in.

p=pressure at entrance, lb. sq. in.

t=temperature of air at entrance, °F.

Then the theoretical discharge is given by :—

$$w^* = \frac{\cdot 53pa}{\sqrt{t+460}}$$

Or, if c=coefficient of discharge for the nozzle.

$$\text{Actual discharge} = w = \frac{\cdot 53cpa}{\sqrt{t+460}}.$$

Thus, if $c = \cdot 98$ and $p = 14\cdot 7$ lb. sq. in.

$$w = \frac{7\cdot 65a}{\sqrt{t+460}}.$$

Air-pump Capacity.—The effective capacity of a reciprocating air-pump depends not only upon the displacement

* For explanation of this formula refer to any standard work on thermodynamics.

of the buckets, but also upon the volumetric efficiency of the pump. The volumetric efficiency refers to the ratio of the actual volume of air taken in per suction stroke, at the partial pressure and temperature obtaining at the air-pump suction, to the suction stroke volume of the pump. This efficiency, however, varies so much with the pressure and temperature, and with the speed, design, and condition of the pump, that it is practically impossible to specify values of volumetric efficiency with any degree of certainty. For ordinary conditions of operation with an Edwards air pump it may be taken to be about 50% at $3\frac{1}{2}$ in. absolute pressure, decreasing to about 18% or less at 1 in. To deal with a definite weight of air the necessary effective displacement of the pump would depend largely upon the partial pressures of the air and the associated vapour at the pump suction, and may be estimated by means of Fig. 39, p. 103, if the conditions are specified. Thus, for example, taking a leakage of ·5 lb. of air per 1000 lb. of steam condensed; vacuum 28 in. (barometer 30 in.), and temperature at air-pump suction 80° F., reference to Fig. 39 shows that the volume of 1 lb. air under these conditions is about 430 cub. ft.

Then, $\frac{\cdot 5 \times 430}{1000} = \cdot 215$ cub. ft. per lb. of steam condensed is the required effective displacement according to the above data. Assuming a volumetric efficiency of, say, ·3, the suction displacement of the air-pump buckets would be

$$\frac{\cdot 215}{\cdot 3} = \cdot 72 \text{ cub. ft. per lb. steam condensed.}$$

A reference to Fig. 69, p. 154, for Professor Weighton's experiments indicates that for reciprocating engines in fairly good condition, and using an efficient type of surface condenser, very little advantage is obtained by increasing the air-pump displacement above ·7 cub. ft. per lb. of steam condensed at full load.

For the highest vacua it may be necessary to water seal all valve spindles in the exhaust line. On the Continent the exhaust pipe joints have also been water-sealed in some installations, but this is hardly necessary if the joints are well

designed and well made. If the exhaust pipe is long some allowance would be necessary in the size of air-pump for the extra air leakage that would occur. Some data regarding the leakage of air is given on p. 111.

Resistance to Flow of Steam through Surface Condensers.—During the experiments referred to on p. 110, Mr. Orrok also measured the increase in vacuum from the condenser inlet to the air-pump suction for several condensers and at various loads. The results obtained are illustrated in Fig. 47 on a load base, though, unfortunately, no plotted points were given in the figure, and no particulars were given respecting the condenser arrangement and the diameter and spacing of the tubes.

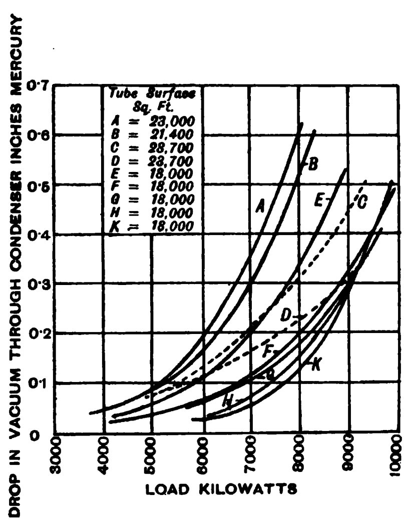


FIG. 47.—Difference of vacuum between inlet and outlet of surface condensers.

In his experiments on the various condensers described on p. 149, and illustrated in Figs. 66 and 67, pp. 150 and 151, Professor Weighton measured the vacuum both at the top and bottom of the condenser, two vacuum gauges and a mercury column being used, all of which had been calibrated from a standard instrument, which in its turn had been standardised in the National Physical Laboratory. Care was taken that no such thing as "induced" reading should occur, due to the current of steam and air, by fitting a curved projecting nozzle to the gauge pipe inside the condenser, and turning it round in all directions and observing its effect during the tests. In no case was the slightest induced effect observed. The vacuum, and therefore the absolute pressure, in these condensers was found to be practically uniform throughout the condenser. In fact, in most tests, the absolute pressure at the bottom was found to be slightly greater than at the top. At first sight this

might seem to be an impossible condition since the tubes will offer some resistance to the flow through the condenser. It has to be remembered, however, that the conditions on the steam side of the tubes are extremely complicated. In the first place the atmospheric pressure at the bottom of the condenser is slightly greater than at the top by the column of air between the two levels; this, however, has only a very small influence on the readings of the vacuum gauges. Secondly, the mixture of steam and air enters the condenser at a fairly high velocity and therefore has a corresponding momentum, whilst the mixture leaves the tubes with a comparatively low momentum. Therefore some of the kinetic energy of flow at the condenser inlet may become converted into pressure energy and thus raise the absolute pressure above the value it would otherwise attain. In ordinary practice, however, the vacuum at the air pump suction may be 0·2 or 0·3 in. of mercury greater than at the condenser inlet.

Without further experiments made with due regard to accuracy it is hardly worth while discussing any probable law of resistance on the steam side of the tubes.

Rate of Heat Transmission in Surface Condensers.—The design of surface condensers is fundamentally based upon the rate of heat transmission between the condensing steam and the cooling or circulating water. As shown previously, there are several factors affecting the resistance to the heat-flow, and the total resistance is the sum of the separate resistances taken in series. The most influential factors affecting the resistance on the steam side of the tubes are: The amount of air present with the steam, the design of the condenser with respect to the velocity of flow, or, the elimination of stagnant spaces, and the cleanliness of the surface of the tubes. On the water side the velocity of the water and the cleanliness of the tubes are important. The resistance of the tubes depends upon the material and the thickness.

Discussing the problem of condensation first as a whole with respect to the rate of heat transmission between the steam and the water.

Let H = heat transmitted to water, B.Th.U. per sec.

h = rate of heat transmission. B.Th.U. per sq. ft. steam side, per sec., per deg. Fahr. steam to water.

A = condensing surface, sq. ft.

w = water-flow, lb. per sec.

T_s = steam inlet temperature, °F.

t_1 = water inlet temperature, °F.

t_2 = ,, outlet ,, °F.

t_m = mean temperature difference, steam to water, °F.

On the steam side it is practically impossible to define the mean temperature of the steam when air is present, because the steam temperature then falls from inlet to outlet in the presence of air, and the rate of fall at any point in the condenser is rarely known. Without air present, however, the mean steam temperature should be practically constant at T_s, and as this represents the ideal conditions, the mean difference of temperature will be taken to be the difference between the steam inlet temperature and the mean temperature of the water, although it is known this does not represent the true mean difference of temperature in most cases.

Therefore, from p. 28

$$t_m = \frac{t_2 - t_1}{\log_e \frac{T_s - t_1}{T_s - t_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$H = w(t_2 - t_1)$$

$$h = \frac{H}{At_m}$$

$$= \frac{w}{A} \log_e \frac{T_s - t_1}{T_s - t_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

This expression (2) will be used generally to calculate the rate of heat transmission h from the experimental data obtained from surface condensers.

The Influence of the Velocity of Flow on the Rate of Heat Transmission from Steam to Water.—This will first be considered as a single problem without attempting to separate the influence of the velocity of the water from the

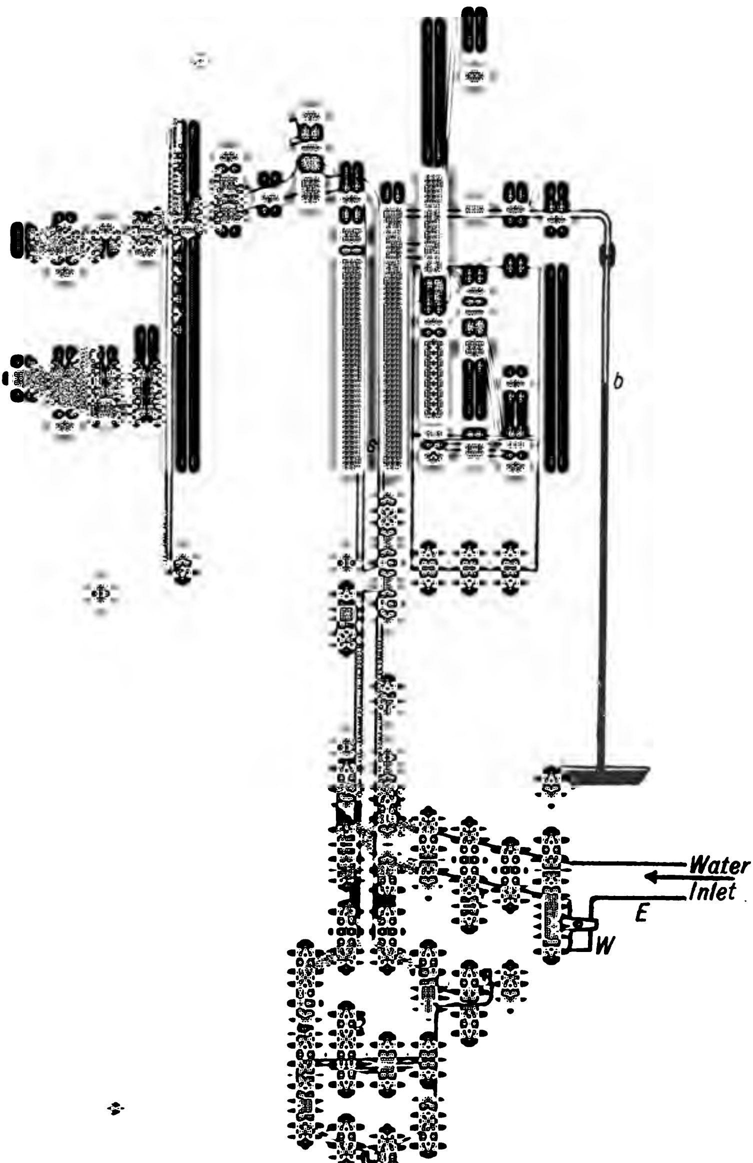
other factors affecting the rate of heat transmission. Also, the results of experiments on small apparatus with single tubes will first be dealt with, leaving the more complicated problems associated with ordinary surface condensers for later discussion. The experiments and the apparatus used will be described in some detail to show the relative value of the particular experiments discussed.

The late Dr. Joule was an early experimenter on the condensation of steam.* To understand the results obtained by him it is necessary to describe the apparatus used and the method of experimenting. The apparatus is shown diagrammatically in Fig. 48. B is the boiler, with steam pipe P and stop cock T, connected to the vertical condensing tube S, at the lower end of which is the receiver R arranged to receive the water of condensation. This receiver had a screwed stopper *n* and a rubber stopper *r*. Cooling water was supplied to the outside of the tube, flowing through the annular space as shown, and collected in the vessel V, previously being allowed to flow into the small can U containing a thermometer. The branch pipe *p* connected the upper end of the tube S to the mercury vacuum column *b*. The rubber connections at *t* and *q* prevented end conduction of heat and the whole apparatus was well insulated by a thick coating of cotton wool and flannel to prevent excessive loss of heat externally.

Every precaution was taken to prevent ingress of air to the tube. In some experiments another vacuum column was connected up to the receiver R at the point *r* and placed side by side with the gauge *b* in the same dish of mercury. Observations were made during rapid and slow condensation and at various vacua, but the height of the columns generally appeared to be exactly the same; if any difference could be observed at any time, the gauge connected to *r* indicated the less perfect vacuum of the two, the difference, however, amounted in no case to more than $\frac{1}{30}$ in.

The method of experimenting was as follows: The nut *n* was unscrewed and the dish of mercury removed from under the gauge tube *b*, all the water was discharged at the tap W

* *Trans. Roy. Soc.*, 1861, Vol. 151.



experiments

and the steam cock T partly opened allowing the steam to blow through the tube S, receiver R, and gauge tube b so as to free them completely from air. The nut n was then screwed on, cock W closed and the water turned on, all as near simultaneously as possible. At the moment when steam was about to cease issuing from the tube b its end was introduced below the mercury level. Condensation then went on with perfect regularity and the vacuum remained constant. The experiment terminated by shutting off the steam and water, opening

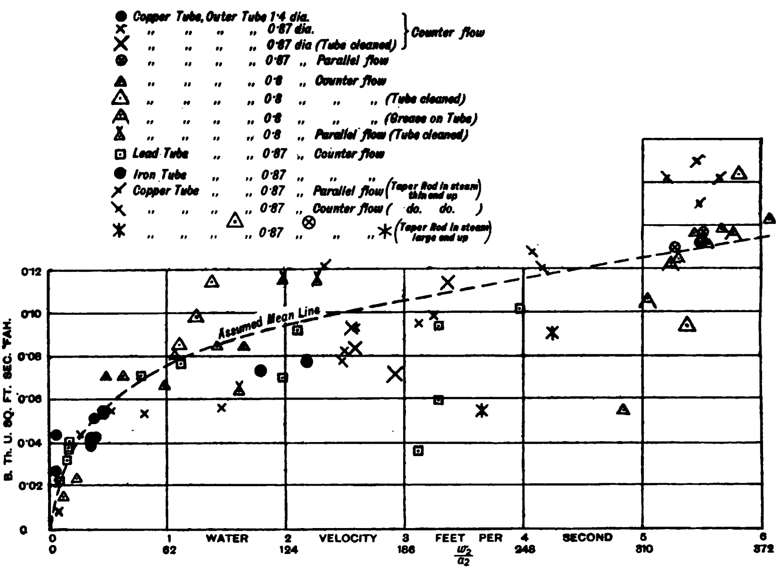


FIG. 49.—Rate of heat transmission from Dr. Joule's condenser experiments.

the cock W, removing the nut n to release the water of condensation, caught in a small can held close and containing a thermometer and then overflowing into a larger vessel, which was then weighed. Small allowances were made for losses by evaporation and by radiation.

It was found that by blowing off steam from the boiler B for ten minutes or so at the commencement of each day's experiments all the air from the water was expelled, and that even if condensation occurred until the receiver R was entirely

filled with water, no change took place in the height of the vacuum gauge.

Numerous experiments were made under a variety of conditions of vacuum, temperature, and rate of water-flow with copper, lead, and iron tubes. The rate of heat transmission

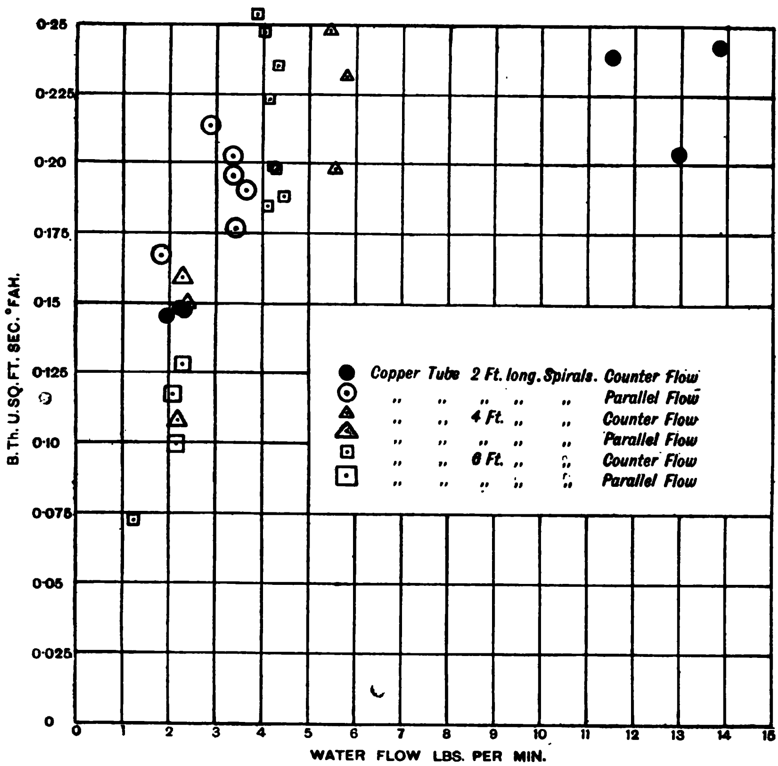


FIG. 50.—Rate of heat transmission from Dr. Joule's condenser experiments.

is shown plotted in Fig. 49 on a base of water velocity, but, unfortunately, the temperatures of the steam and of the water of condensation were different for each separate experiment. The values of the rate of heat transmission were deduced by equation 2, p. 121, and were based on the average of the inner and outer areas of the tube. It will be noticed how erratic the results appear to be, probably being

mostly due to the erratic conditions on the steam side of the tube. In a few of these experiments a tapered rod was placed inside the steam tube.

A number of experiments were also made with various tubes having spirals of copper wire round the tube to break up the flow of water. Some of the results are shown in Figs. 50 and 51 plotted on a base of water-flow in pounds per minute.

In the original paper the results are only shown in tabular form and from an inspection of the values Dr. Joule made the

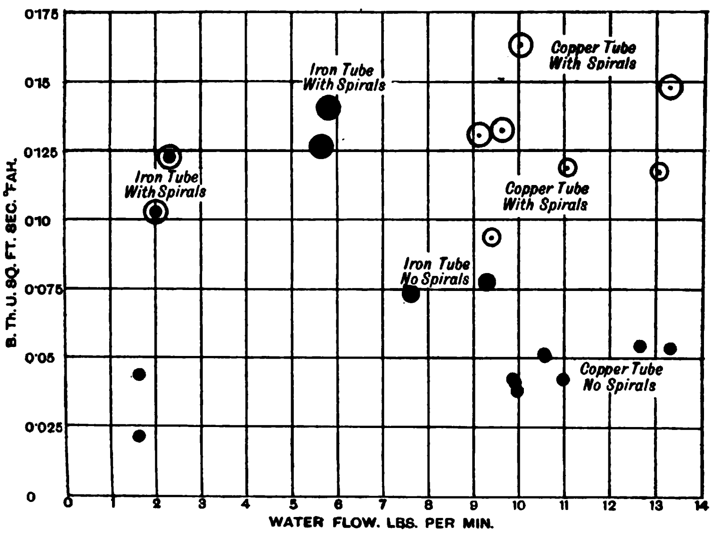


FIG. 51.—Rate of heat transmission from Dr. Joule's condenser experiments.

following deductions (among others) which are more or less confirmed by the graphs in Figs. 49 to 51 :—

1. The pressure in the vacuous space was sensibly equal in all parts.

2. In this arrangement it is a matter of indifference in which direction the water is transmitted. Hence,

3. The temperature of the vacuous space was sensibly equal in all parts.

4. The rate of heat transmission was little influenced by the kind of metal for the tubes or by its thickness for ordinary

limits, or even by the state of its surface as to greasiness or oxidation.

5. Narrowing the steam space by placing a rod in the axis of the steam tube did not produce any sensible effect.

6. By means of a contrivance for the automatical agitation of the particles of the refrigerating stream, such as the spirals employed, an improvement in the rate of heat transmission for a given head of water takes place.

Owing to the erratic nature of the results relating to Dr. Joule's experiments the writer does not think it worth while to show the values of M derived from equation 5, p. 29, where T_1 and T_2 are inlet and outlet water temperatures respectively.

Another early experimenter who called attention to the influence of the velocity of the cooling water was B. G. Nichol.* His apparatus consisted of a brass tube ¾ in. external diameter, of No. 18 B.W.G. thickness, enclosed in a wrought-iron pipe 3¾ in. diameter and length 5 ft. 5⅛ in. between the wrought iron ends. Indiarubber glands were used to keep the ends of the pipe tight around the tube. The cooling water entered the brass tube at about 58° F. and the steam condensed at about 225° F. on the outside of this tube, the water of condensation leaving at about 200° F. The whole apparatus was well lagged to prevent external losses of heat. Three tests were recorded with the tube vertical and three with it horizontal. The increase of the rate of heat transmission (calculated by equation 2, p. 121) with the velocity is shown in Fig. 52.

Using equation 5, p. 29, the calculated values of M are also shown plotted in Fig. 52.

Another experimenter on the transmission of heat in a single tube condenser was G. A. Hagemann, of Copenhagen.† His final apparatus consisted of a brass tube 2 mm. (·079 in.) thick, and 49 mm. (1·93 in.) outside diameter, having a length of 941 mm. (37·1 in.). The brass tube was fitted steam tight in the centre of a cast-iron pipe, about 6 in. diameter inside,

* *Engineering*, 10th Dec., 1875.

† "Experiments on the Transmission of Heat," *Proc. I.C.E.*, Vol. LXXVII, 1883–84, Part III.

of the buckets, but also upon the volumetric efficiency of the pump. The volumetric efficiency refers to the ratio of the actual volume of air taken in per suction stroke, at the partial pressure and temperature obtaining at the air-pump suction, to the suction stroke volume of the pump. This efficiency, however, varies so much with the pressure and temperature, and with the speed, design, and condition of the pump, that it is practically impossible to specify values of volumetric efficiency with any degree of certainty. For ordinary conditions of operation with an Edwards air pump it may be taken to be about 50% at $3\frac{1}{2}$ in. absolute pressure, decreasing to about 18% or less at 1 in. To deal with a definite weight of air the necessary effective displacement of the pump would depend largely upon the partial pressures of the air and the associated vapour at the pump suction, and may be estimated by means of Fig. 39, p. 103, if the conditions are specified. Thus, for example, taking a leakage of ·5 lb. of air per 1000 lb. of steam condensed; vacuum 28 in. (barometer 30 in.), and temperature at air-pump suction 80° F., reference to Fig. 39 shows that the volume of 1 lb. air under these conditions is about 430 cub. ft.

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$$\frac{\cdot 215}{\cdot 3} = \cdot 72 \text{ cub. ft. per lb. steam condensed.}$$

A reference to Fig. 69, p. 154, for Professor Weighton's experiments indicates that for reciprocating engines in fairly good condition, and using an efficient type of surface condenser, very little advantage is obtained by increasing the air-pump displacement above ·7 cub. ft. per lb. of steam condensed at full load.

For the highest vacua it may be necessary to water seal all valve spindles in the exhaust line. On the Continent the exhaust pipe joints have also been water-sealed in some installations, but this is hardly necessary if the joints are well

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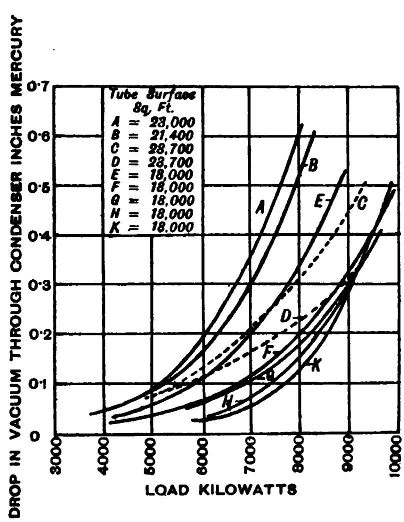

FIG. 47.—Difference of vacuum between inlet and outlet of surface condensers.

In his experiments on the various condensers described on p. 149, and illustrated in Figs. 66 and 67, pp. 150 and 151, Professor Weighton measured the vacuum both at the top and bottom of the condenser, two vacuum gauges and a mercury column being used, all of which had been calibrated from a standard instrument, which in its turn had been standardised in the National Physical Laboratory. Care was taken that no such thing as "induced" reading should occur, due to the current of steam and air, by fitting a curved projecting nozzle to the gauge pipe inside the condenser, and turning it round in all directions and observing its effect during the tests. In no case was the slightest induced effect observed. The vacuum, and therefore the absolute pressure, in these condensers was found to be practically uniform throughout the condenser. In fact, in most tests, the absolute pressure at the bottom was found to be slightly greater than at the top. At first sight this

might seem to be an impossible condition since the tubes will offer some resistance to the flow through the condenser. It has to be remembered, however, that the conditions on the steam side of the tubes are extremely complicated. In the first place the atmospheric pressure at the bottom of the condenser is slightly greater than at the top by the column of air between the two levels; this, however, has only a very small influence on the readings of the vacuum gauges. Secondly, the mixture of steam and air enters the condenser at a fairly high velocity and therefore has a corresponding momentum, whilst the mixture leaves the tubes with a comparatively low momentum. Therefore some of the kinetic energy of flow at the condenser inlet may become converted into pressure energy and thus raise the absolute pressure above the value it would otherwise attain. In ordinary practice, however, the vacuum at the air pump suction may be 0·2 or 0·3 in. of mercury greater than at the condenser inlet.

Without further experiments made with due regard to accuracy it is hardly worth while discussing any probable law of resistance on the steam side of the tubes.

Rate of Heat Transmission in Surface Condensers.—The design of surface condensers is fundamentally based upon the rate of heat transmission between the condensing steam and the cooling or circulating water. As shown previously, there are several factors affecting the resistance to the heat-flow, and the total resistance is the sum of the separate resistances taken in series. The most influential factors affecting the resistance on the steam side of the tubes are: The amount of air present with the steam, the design of the condenser with respect to the velocity of flow, or, the elimination of stagnant spaces, and the cleanliness of the surface of the tubes. On the water side the velocity of the water and the cleanliness of the tubes are important. The resistance of the tubes depends upon the material and the thickness.

Discussing the problem of condensation first as a whole with respect to the rate of heat transmission between the steam and the water.

Let H=heat transmitted to water, B.Th.U. per sec.

h=rate of heat transmission. B.Th.U. per sq. ft. steam side, per sec., per deg. Fahr. steam to water.

A=condensing surface, sq. ft.

w=water-flow, lb. per sec.

T_s=steam inlet temperature, °F.

t_1=water inlet temperature, °F.

t_2= ,, outlet ,, °F.

t_m=mean temperature difference, steam to water, °F.

On the steam side it is practically impossible to define the mean temperature of the steam when air is present, because the steam temperature then falls from inlet to outlet in the presence of air, and the rate of fall at any point in the condenser is rarely known. Without air present, however, the mean steam temperature should be practically constant at T_s, and as this represents the ideal conditions, the mean difference of temperature will be taken to be the difference between the steam inlet temperature and the mean temperature of the water, although it is known this does not represent the true mean difference of temperature in most cases.

Therefore, from p. 28

$$t_m = \frac{t_2 - t_1}{\log_e \dfrac{T_s - t_1}{T_s - t_2}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

$$H = w(t_2 - t_1)$$

$$h = \frac{H}{At_m}$$

$$= \frac{w}{A} \log_e \frac{T_s - t_1}{T_s - t_2} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

This expression (2) will be used generally to calculate the rate of heat transmission h from the experimental data obtained from surface condensers.

The Influence of the Velocity of Flow on the Rate of Heat Transmission from Steam to Water.—This will first be considered as a single problem without attempting to separate the influence of the velocity of the water from the

other factors affecting the rate of heat transmission. Also, the results of experiments on small apparatus with single tubes will first be dealt with, leaving the more complicated problems associated with ordinary surface condensers for later discussion. The experiments and the apparatus used will be described in some detail to show the relative value of the particular experiments discussed.

The late Dr. Joule was an early experimenter on the condensation of steam.* To understand the results obtained by him it is necessary to describe the apparatus used and the method of experimenting. The apparatus is shown diagrammatically in Fig. 48. B is the boiler, with steam pipe P and stop cock T, connected to the vertical condensing tube S, at the lower end of which is the receiver R arranged to receive the water of condensation. This receiver had a screwed stopper *n* and a rubber stopper *r*. Cooling water was supplied to the outside of the tube, flowing through the annular space as shown, and collected in the vessel V, previously being allowed to flow into the small can U containing a thermometer. The branch pipe *p* connected the upper end of the tube S to the mercury vacuum column *b*. The rubber connections at *t* and *q* prevented end conduction of heat and the whole apparatus was well insulated by a thick coating of cotton wool and flannel to prevent excessive loss of heat externally.

Every precaution was taken to prevent ingress of air to the tube. In some experiments another vacuum column was connected up to the receiver R at the point *r* and placed side by side with the gauge *b* in the same dish of mercury. Observations were made during rapid and slow condensation and at various vacua, but the height of the columns generally appeared to be exactly the same; if any difference could be observed at any time, the gauge connected to *r* indicated the less perfect vacuum of the two, the difference, however, amounted in no case to more than $\frac{1}{30}$ in.

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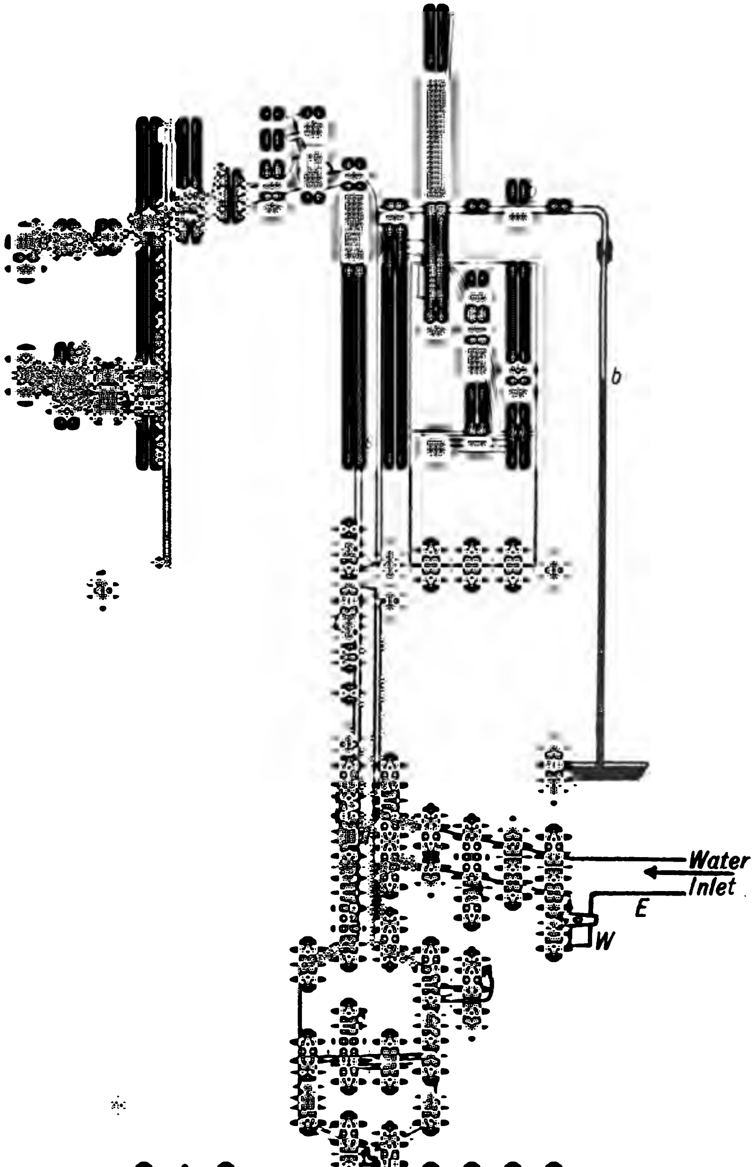

[illegible] experiments [illegible] steam.

and the steam cock T partly opened allowing the steam to blow through the tube S, receiver R, and gauge tube b so as to free them completely from air. The nut n was then screwed on, cock W closed and the water turned on, all as near simultaneously as possible. At the moment when steam was about to cease issuing from the tube b its end was introduced below the mercury level. Condensation then went on with perfect regularity and the vacuum remained constant. The experiment terminated by shutting off the steam and water, opening

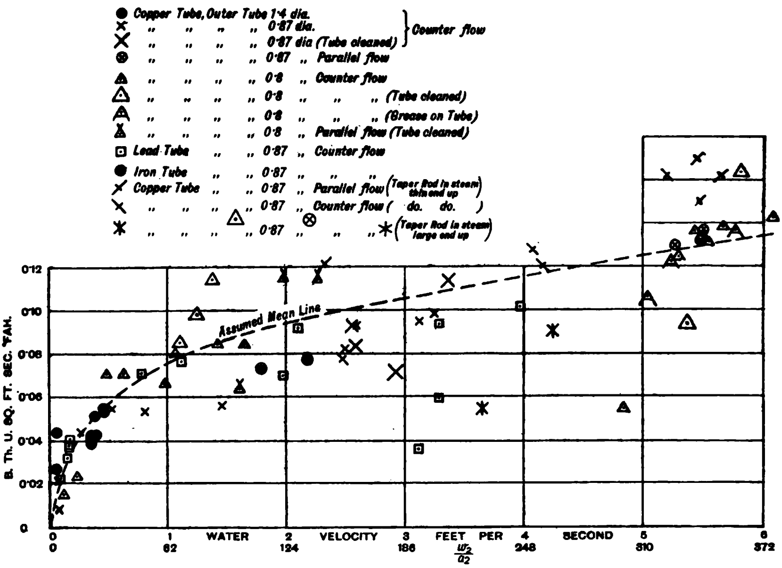

FIG. 49.—Rate of heat transmission from Dr. Joule's condenser experiments.

the cock W, removing the nut n to release the water of condensation, caught in a small can held close and containing a thermometer and then overflowing into a larger vessel, which was then weighed. Small allowances were made for losses by evaporation and by radiation.

It was found that by blowing off steam from the boiler B for ten minutes or so at the commencement of each day's experiments all the air from the water was expelled, and that even if condensation occurred until the receiver R was entirely

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Numerous experiments were made under a variety of conditions of vacuum, temperature, and rate of water-flow with copper, lead, and iron tubes. The rate of heat transmission

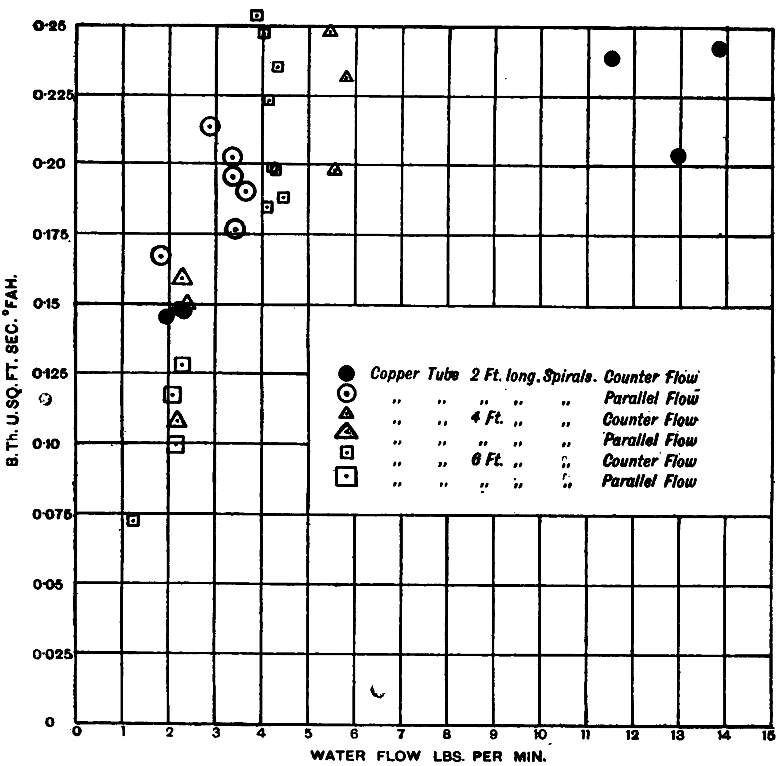

FIG. 50.—Rate of heat transmission from Dr. Joule's condenser experiments.

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A number of experiments were also made with various tubes having spirals of copper wire round the tube to break up the flow of water. Some of the results are shown in Figs. 50 and 51 plotted on a base of water-flow in pounds per minute.

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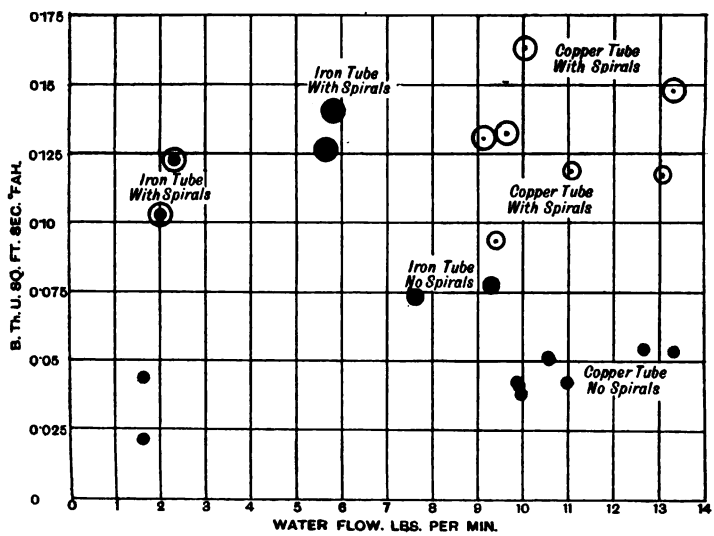

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following deductions (among others) which are more or less confirmed by the graphs in Figs. 49 to 51 :—

1. The pressure in the vacuous space was sensibly equal in all parts.

2. In this arrangement it is a matter of indifference in which direction the water is transmitted. Hence,

3. The temperature of the vacuous space was sensibly equal in all parts.

4. The rate of heat transmission was little influenced by the kind of metal for the tubes or by its thickness for ordinary

limits, or even by the state of its surface as to greasiness or oxidation.

5. Narrowing the steam space by placing a rod in the axis of the steam tube did not produce any sensible effect.

6. By means of a contrivance for the automatical agitation of the particles of the refrigerating stream, such as the spirals employed, an improvement in the rate of heat transmission for a given head of water takes place.

Owing to the erratic nature of the results relating to Dr. Joule's experiments the writer does not think it worth while to show the values of M derived from equation 5, p. 29, where T_1 and T_2 are inlet and outlet water temperatures respectively.

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Using equation 5, p. 29, the calculated values of M are also shown plotted in Fig. 52.

Another experimenter on the transmission of heat in a single tube condenser was G. A. Hagemann, of Copenhagen.† His final apparatus consisted of a brass tube 2 mm. (·079 in.) thick, and 49 mm. (1·93 in.) outside diameter, having a length of 941 mm. (37·1 in.). The brass tube was fitted steam tight in the centre of a cast-iron pipe, about 6 in. diameter inside,

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of the buckets, but also upon the volumetric efficiency of the pump. The volumetric efficiency refers to the ratio of the actual volume of air taken in per suction stroke, at the partial pressure and temperature obtaining at the air-pump suction, to the suction stroke volume of the pump. This efficiency, however, varies so much with the pressure and temperature, and with the speed, design, and condition of the pump, that it is practically impossible to specify values of volumetric efficiency with any degree of certainty. For ordinary conditions of operation with an Edwards air pump it may be taken to be about 50% at $3\frac{1}{2}$ in. absolute pressure, decreasing to about 18% or less at 1 in. To deal with a definite weight of air the necessary effective displacement of the pump would depend largely upon the partial pressures of the air and the associated vapour at the pump suction, and may be estimated by means of Fig. 39, p. 103, if the conditions are specified. Thus, for example, taking a leakage of ·5 lb. of air per 1000 lb. of steam condensed; vacuum 28 in. (barometer 30 in.), and temperature at air-pump suction 80° F., reference to Fig. 39 shows that the volume of 1 lb. air under these conditions is about 430 cub. ft.

Then, $\frac{\cdot 5 \times 430}{1000} = \cdot 215$ cub. ft. per lb. of steam condensed is the required effective displacement according to the above data. Assuming a volumetric efficiency of, say, ·3, the suction displacement of the air-pump buckets would be

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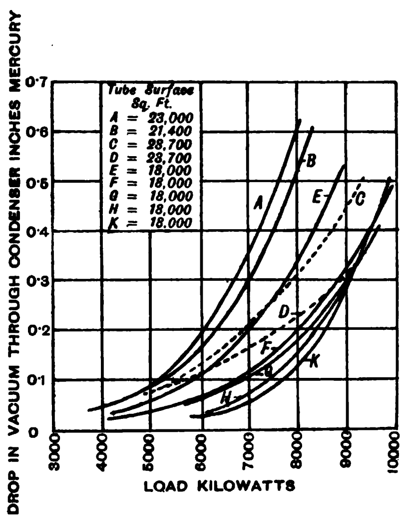

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h = rate of heat transmission. B.Th.U. per sq. ft. steam side, per sec., per deg. Fahr. steam to water.

A = condensing surface, sq. ft.

w = water-flow, lb. per sec.

T_s = steam inlet temperature, °F.

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$$H = w(t_2 - t_1)$$

$$h = \frac{H}{A t_m}$$

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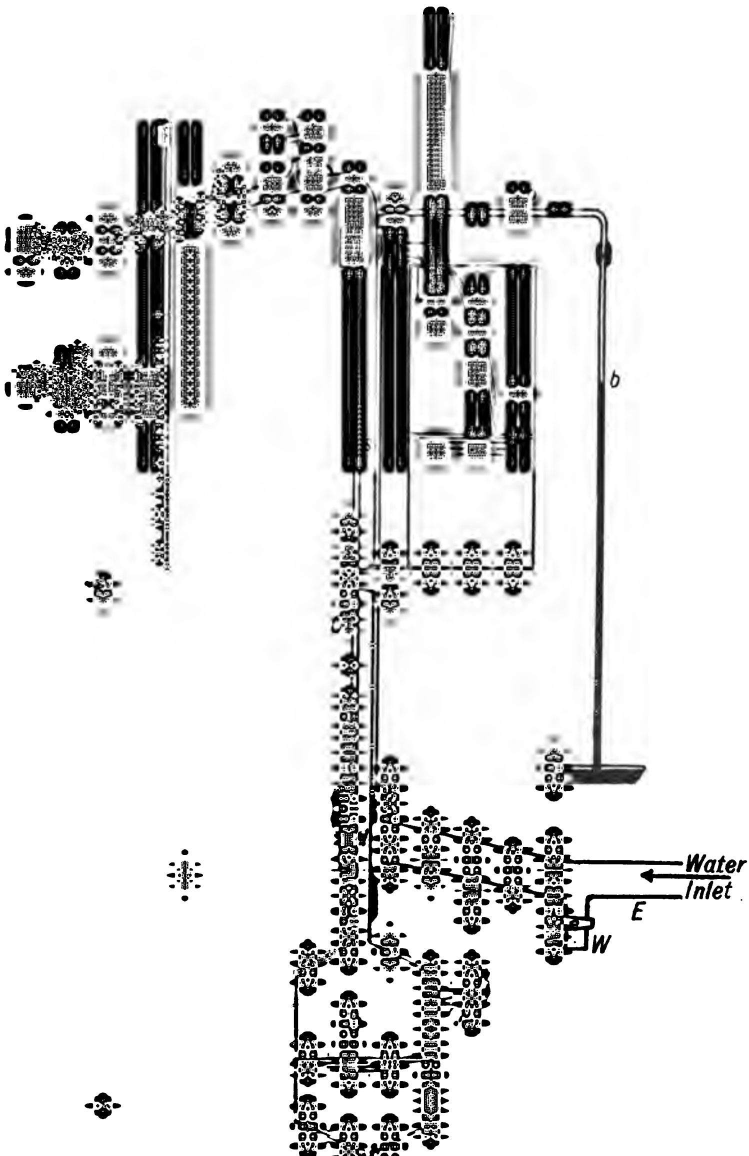

[illegible] experiments [illegible]

and the steam cock T partly opened allowing the steam to blow through the tube S, receiver R, and gauge tube *b* so as to free them completely from air. The nut *n* was then screwed on, cock W closed and the water turned on, all as near simultaneously as possible. At the moment when steam was about to cease issuing from the tube *b* its end was introduced below the mercury level. Condensation then went on with perfect regularity and the vacuum remained constant. The experiment terminated by shutting off the steam and water, opening

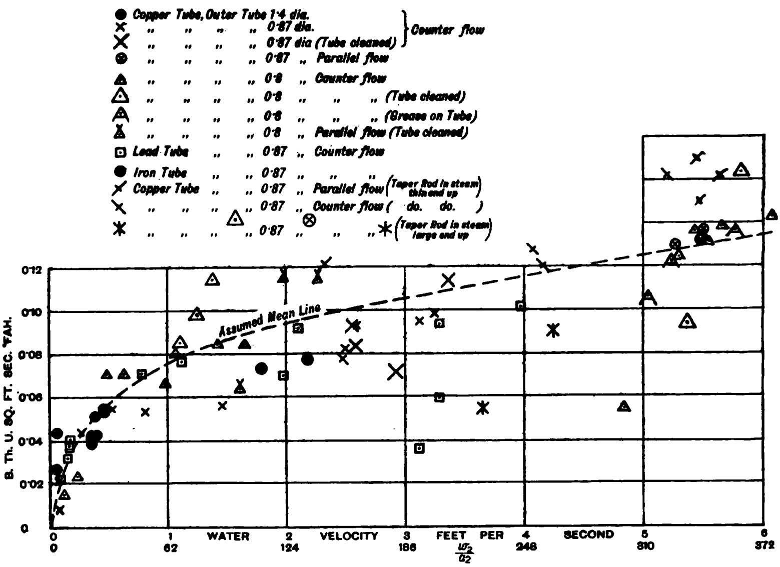

FIG. 49.—Rate of heat transmission from Dr. Joule's condenser experiments.

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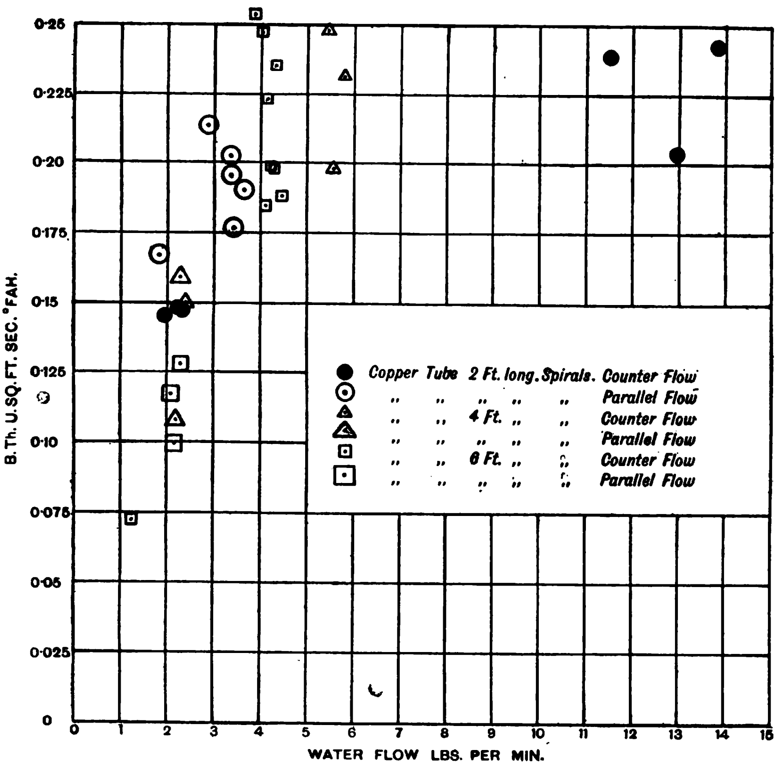

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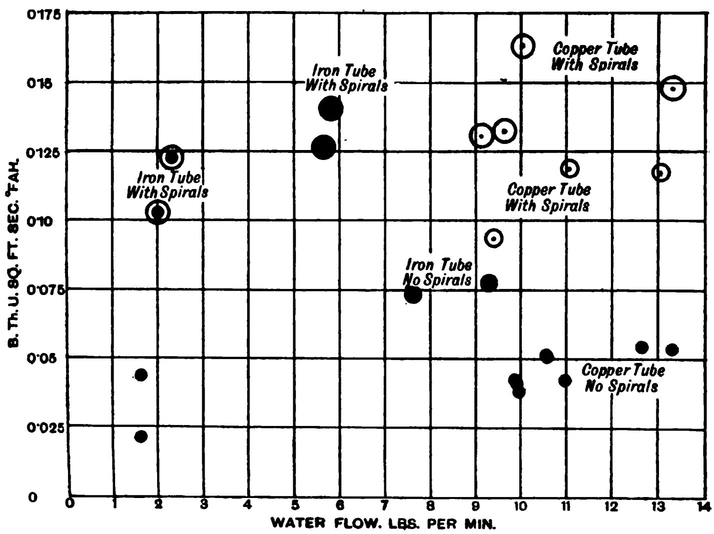

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5. Narrowing the steam space by placing a rod in the axis of the steam tube did not produce any sensible effect.

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Owing to the erratic nature of the results relating to Dr. Joule's experiments the writer does not think it worth while to show the values of M derived from equation 5, p. 29, where T_1 and T_2 are inlet and outlet water temperatures respectively.

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the steam condensed on the outside of the brass tube and the cooling water flowed through. The lower end of the brass

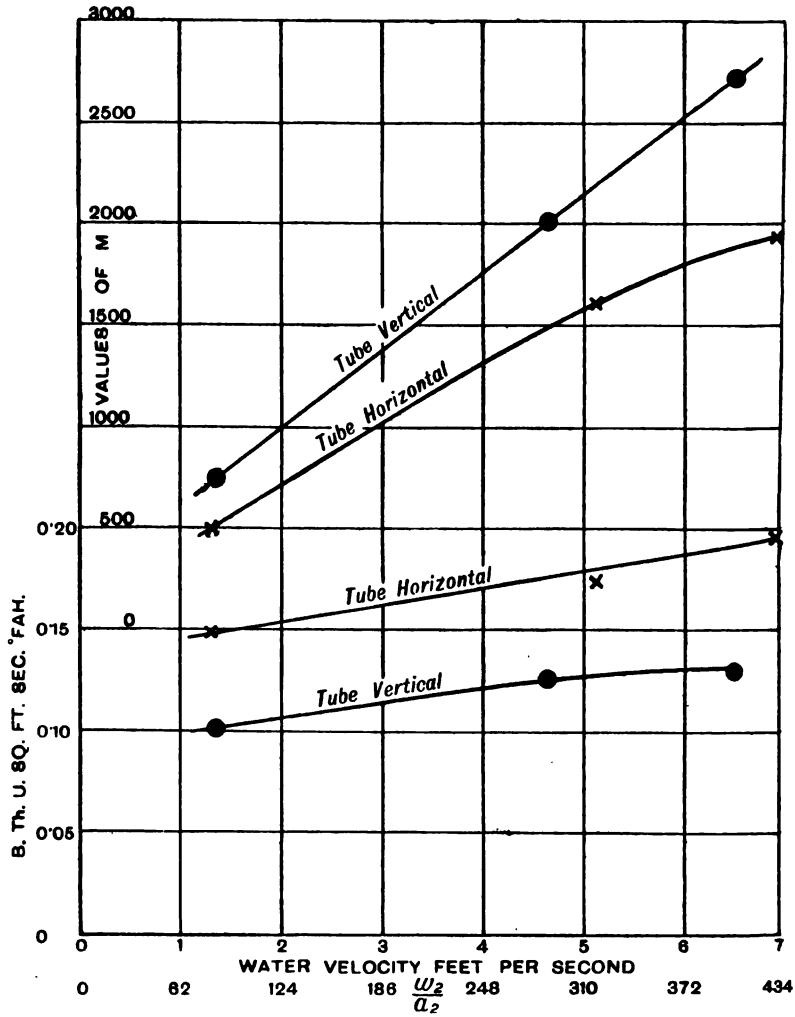


FIG. 52.—Rate of heat transmission from Nichol's condenser experiments.

tube communicated with a water supply tank in which was placed a copper steam coil whereby the water could be raised

to any desired temperature before entering the tube. It was soon found necessary to introduce a blocked tube, 38·5 mm. (1·51 in.) diameter externally, inside the brass tube, the water then flowing through the annular space. Without this the temperature of the heated water could not be measured accurately on account of the wide bore and the low water velocities used allowing local differences of temperature of the water passing the thermometer, that is, in other words, the stream of water did not mix completely in passing through the tube. With the block tube in position there was little or no difficulty in this respect. The inlet and outlet temperatures of the water were measured by means of delicate thermometers.

A series of tests were made with the steam inlet temperature at about 212° F. and with the inlet water at various temperatures. Calculating the rate of heat transmission by means of equation 2, p. 121, the various values obtained are shown plotted in Fig. 53 on the base of water velocity. The circles refer to the tests with the steam inlet at 212° F., the inlet temperature of the water being marked against the points, and it would be noticed that the results are somewhat erratic, but that there is a general tendency for the rate of heat transmission to increase with the inlet temperature of the water. Unfortunately, the outlet temperature of the water of condensation does not appear to have been measured.

It is probable that the higher rates of heat transmission with the higher inlet water temperatures were mostly due to the conditions on the steam side of the tube, though some of it might be due to the lower viscosity of the water as suggested previously, and therefore to its greater tendency to eddying motion as the inlet temperature of the water was increased.

The results shown in Fig. 53 as crosses were made with various steam inlet temperatures indicated opposite the various points, the inlet water temperatures being kept at about 44° F. While there was apparently some indication that the rate of heat transmission from the steam to the water increased with the temperature of the steam, no satisfactory conclusion could be drawn from these experiments because the outlet temperatures of the water of condensation

were not recorded. There is a general increase of the rate of heat transmission with an increase of the water velocity. Using equation 5, p. 29, the values of M are also shown plotted in Fig. 53.

Professor L. Ser described some experiments he had made

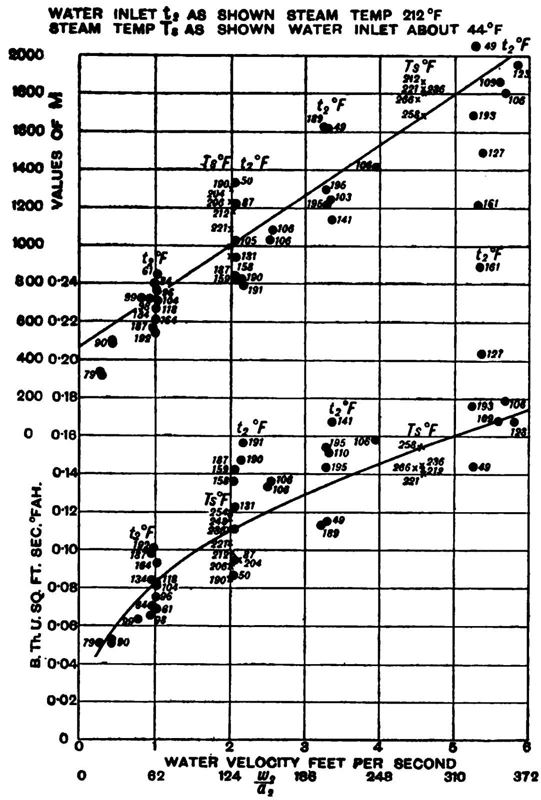


FIG. 53.—Rate of heat transmission from Hagemann's condenser experiments.

on the transmission of heat from a surface to water and from steam to water in his book on *Traité de Physique Industrielle*, Vol. I, 1888. The apparatus used was similar to that described previously, except that the annular area through which the warm water flowed was made equal to the area of the tube in which flowed the cold water. In this case it was arranged to have the

velocities of the warm and cold water the same, and Professor Ser then assumed that the temperature of the tube would be a mean between that of the water on the two sides. The results tabulated by him have been converted and plotted in Fig. 54 as curve (1).

A similar series of experiments were made with steam at 100° C. in the annular space, with water flowing through the tube as in ordinary condenser practice. The tube of copper was 10 mm. (·39 in.) inside diameter, 1 mm. thick, and 314 mm. long. Again from the tabulated values the results shown by curve (2) in Fig. 54 were derived.

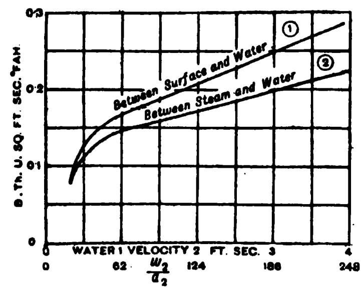


Fig. 54.—Rate of heat transmission from Ser's condenser experiments.

Considering now experiments of comparatively recent date, Fig. 76, p. 162, shows the rate of heat transmission, plotted on a base of water velocity, obtained by J. A. Smith with the apparatus illustrated in Fig. 73, p. 159. In the series A the condensing steam was nearly free from air and thus the ratio $\frac{\text{Steam pressure}}{\text{Total pressure}}=\frac{P_s}{P_t}=1$ nearly, but in series B, $\frac{P_s}{P_t}=\cdot 922$. In both cases the velocity of the water had a considerable influence on the rate of heat transmission, and the differences between the results for series A and series B show the influence of the air on the steam side of the tube. It should be noted, however, that in this apparatus, Fig. 73, the stagnant conditions on the steam side of the tube may have had a considerable influence on these results.

For this condenser $\frac{l}{m_2}=500$, and Fig. 76 also shows the various values of M obtained by calculation from equation 5, p. 29, using for T_s the recorded steam temperatures in the vessel.

Experiments have been made by the writer in the mechanical engineering laboratory at The Royal Technical College, Glasgow, on a small high-speed condenser. The results illustrate the

influence of the water-flow and the effect of the outlet temperature of the air and water of condensation on the rate of heat transmission. The condenser consisted of two brass tubes, one of $1\frac{5}{8}$ in. internal diameter surrounding the other of $1\frac{1}{2}$ in. external diameter which left an annular space between the tubes $\frac{1}{16}$ in. wide. These tubes were enclosed in a cast-iron shell about $2\frac{1}{2}$ in. diameter and placed horizontally. The total tube condensing surface was 2·5 sq. ft. The steam was taken from a steam supply main and passed first through the inner tube and then to the outside of the outer tube, the inlet steam being usually superheated a few degrees by the throttling action from the steam pipe pressure to the condenser pressure. The cooling water from the town's main was arranged to flow between the two tubes, the length of water path being about 3·1 ft.

The first series of experiments were made in 1910, although the condenser was built in 1906, and the rates of heat transmission obtained are shown by the black points in Fig. 55, the steam inlet saturation temperature being 212° F. and the outlet varying in different sets of tests from about 206° F. to about 124° F. as indicated on the curves. It is seen that the rate of heat transmission is greatly influenced by the temperature at which the air and water of condensation were liberated from the condenser. This rate was calculated from the experimental results by the formula 2, p. 121, taking the steam temperature to be 212° F.

A similar series of experiments were made in 1919 with the steam inlet 212° F. and the outlet a few degrees lower. Between 1910 and 1919 the condenser had been used for students' work for a few weeks during each intervening session, and for the remaining periods the water side of the condenser was kept flooded, but without flow. The 1919 results of the rate of heat transmission with steam condensing at 212° F. are shown by the small circles with central dot, being much lower than the top curve of 1910 with which it is comparable. Experiments were also made in 1919 with steam at 34 lb. and also at 74 lb. per sq. in. absolute, the steam outlet in each case being only a few degrees below the inlet temperature. The rate of heat transmission results with

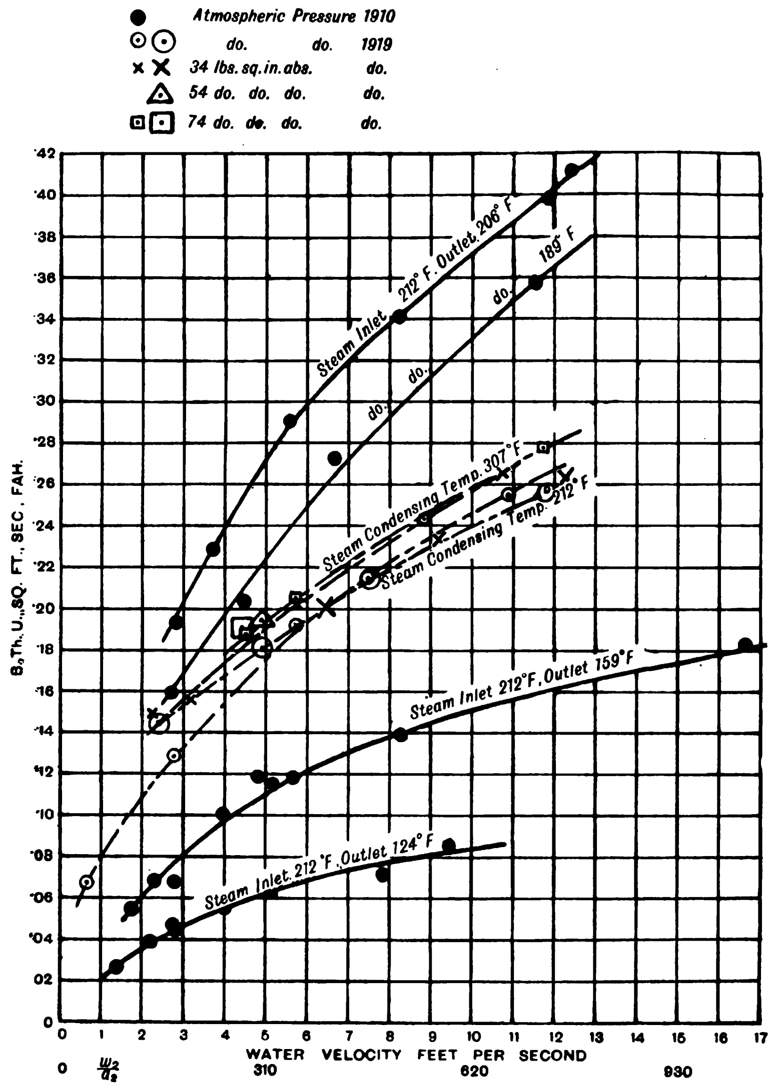


FIG. 55.—Rate of heat transmission in high-speed condenser, from Royds' experiments.

34 lb. pressure are shown by the small crosses, and the 74 lb. by the small square points. The condenser was then dismantled to see whether the difference between the 1910 and the 1919 results were due to accumulations on the tubes. The tubes were found to be covered with a thin deposit of rust, evidently obtained from the steel pipes leading to the condenser. Also the outer tube was found to be cracked along its length, but there had been no signs of leakage at any time during its life, and on breaking the tube it was seen that a very thin strip at the bottom of the crack had remained intact.

To make more certain that leakage at the cracked tube had not caused the difference between the 1919 and the 1910 results, another condenser made to the same patterns was substituted which had been used occasionally for general condensing purposes over the same period without being cleaned. Condensing steam at atmospheric pressure under the same conditions as with the previous condenser gave the results shown by the large circles with central dot, while similar experiments with 34 lb. per sq. in. absolute pressure gave the large crosses. An experiment was also made with 54 lb. per sq. in. absolute pressure shown by the large triangular point and one at 74 lb. shown by the large square point. The results with the new condenser are thus practically in agreement with those obtained in 1919 from the old one. There were no signs of leakage with the new condenser.

An elaborate series of experiments on a single tube condenser have been made by G. A. Orrok,* of New York. The apparatus consisted of a wrought-steel pipe or shell, with cast-iron flanges at each end, drilled and tapped for the various steam, water, and vacuum connections. The condensing tube of 1 in. outside diameter passed through the centre of the shell and the length of the apparatus was such that exactly 1 sq. ft. of outside tube surface was included between the end flanges, the condenser ends being made tight by stuffing boxes where the tube passed through. The circulating water was taken from a salt-water fire supply line and was measured by passing through a calibrated meter, the inlet

* "The Transmission of Heat in Surface Condensation," *Journal Amer. Soc. Mech. Engs.*, Nov., 1910, p. 1773.

and outlet temperatures being measured by thermometers inserted into deep pockets well into the stream of water. The steam for condensation was at first taken from an exhaust main which carried the exhaust steam from various auxiliary engines to an open feed water heater, a cock being used to regulate the flow of steam into the condenser. The inlet temperature of the steam and that of the water of condensation at the outlet were measured, but, unfortunately, the latter temperatures were not recorded in the paper. A connection was made from the condenser to a vacuum line to maintain a constant vacuum in the condenser during any experiment. The water of condensation was allowed to run into a closed calibrated measuring vessel.

With this arrangement it was found impossible to get consistent results, which was ascribed to air leakage and to deposit of oil or dirt on the condensing surface of the tube, as well as to probable variations in the quality of the steam supplied to the condenser. The apparatus was rearranged with a small independent boiler in which steam was generated by a high-pressure steam coil, the water of condensation from the condenser being returned to the boiler, this forming practically a closed circuit, with a make-up supply pipe attached to the boiler to maintain a constant level.

The leakage of air into the condenser was tested before every series of experiments, and it was found that it could be made tight enough to allow only a drop of vacuum from 28 to 27 in. of mercury in fifteen minutes with all valves closed. In order that the condition of the tube as to cleanliness should not affect the results, all the tubes used for testing were thoroughly cleaned every ten or twelve tests.

Tests were made on 1 in. outside diameter Admiralty tubes under various conditions, and also with tubes of the other materials referred to on p. 137. The results obtained with a new 1-in. Admiralty tube with various water velocities are shown in Fig. 56, where the rate of heat transmission has been plotted on a velocity base. In one set the steam inlet temperature was about 126° F. and in another about 187° F., from which it is seen that the steam temperature does not seem to have had any appreciable influence on the rate of

heat transmission. The cooling water inlet was about 50° F. and the rise of temperature obtained was rarely more than a few degrees.

A similar series of tests were made with a 1-in. Admiralty tube taken from an old condenser and which had therefore become corroded to some extent. The results obtained with a steam temperature of about 135° F. are also shown plotted in Fig. 5?, the cooling water inlet temperature being about 50° F. as in the other tests. It will be noted that the rates

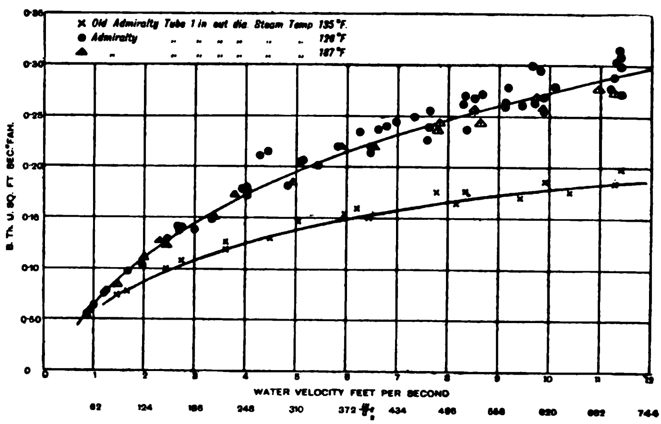


FIG. 56.—Rate of heat transmission from Orrok's condenser experiments.

of heat transmission were appreciably lower than those obtained with the new tube.

The corresponding values of M from equation 5, p. 29, taking T_s as the inlet steam temperature are shown plotted in Fig. 57 using the value of $\frac{l}{m_2}=204$ for this apparatus.

Various other tests were made at constant water velocity with various degrees of vacua as well as some with various temperatures of the inlet water. Although the results obtained may be interesting it is not proposed to describe or to discuss these tests at length because the results will be to some extent dependent upon the outlet temperature of the water of condensation and air and no definite information was given of these values.

The influence of the various tube materials on the rate of heat transmission was summarised by Mr. Orrok, and it would perhaps be worth while to quote this: "Taking the heat transmission for the copper tube as 1, under similar conditions the transfer for other materials was approximately as follows:

Copper 1·0; Admiralty ·98; aluminium lined ·97; Admiralty oxidised (black) ·92; aluminium bronze ·87; cupro-nickel ·8; tin ·79; Admiralty lead lined ·79; zinc ·75; Monel metal ·74; Shelby steel ·63; old Admiralty (badly corroded) ·55; Admiralty (vulcanised inside) ·47; glass ·25; Admiralty (vulcanised both sides) ·17."

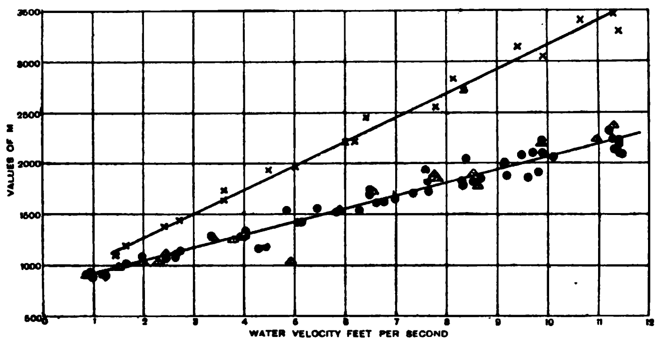


FIG. 57.—Values of "M" from Orrok's condenser experiments.

All the tubes were 1 in. outside diameter, and thickness 18 B.W.G., except the copper tube 16 B.W.G., Admiralty lead lined ·088 in., and tin ·055 in. thick. The velocity of the water was 8·6 ft. per second in the experiments made for the above comparisons.

The above ratios should only be regarded as a measure of the influence of the material of the tube on the total resistance to the flow of heat between the steam and the water under the particular conditions of operation. If, for example, the velocity of the water had been much lower than 8·6 ft. per sec., the resistance to heat-flow between the tube surface and the water would have been correspondingly higher and the resistance of the tube proportionally smaller compared with the total resistance between the steam and the water. In other

words, the lower the velocity of the water the less important is the material and thickness of the tubes.

The illustration in Fig. 58 shows the apparatus used by J. K. Clement and C. M. Garland* working with steam above atmospheric pressure. The apparatus consisted of the steam jacket F having flanges and stuffing boxes at each end through which passed a cold-drawn steel tube, of outside diameter 1·253 in. and inside ·985 in. Each end of the tube had a flange, and a rubber joint ¼ in. thick was inserted to prevent excessive conduction of heat along the tube to the inlet or outlet water-pipe. Temperatures of steam and water were measured as shown in the illustration, baffles being inserted in the water-

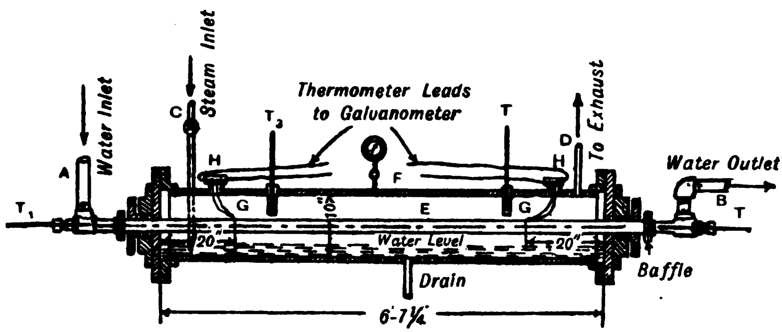


FIG. 58.—Condenser used by Clement and Garland.

pipe at inlet and outlet to break up the flow of water near the thermometers. The water was taken from a constant head tank, located about 25 ft. above the tube, which permitted a maximum velocity of about 17 ft. per sec. The steam was admitted at C and bubbled up through the water in the jacket, which was kept at a constant level, and passed out with the air at the pipe D. The bubbling of the steam through the water got rid of the superheat of the steam due to throttling at the valve C. The steam pressure in the jacket, and therefore the temperature, was maintained practically constant at the desired value by placing a safety valve on the outlet pipe D and allowing the steam to blow through this valve

* "A Study in Heat Transmission," Bulletin No. 4, Sept., 1909, University of Illinois Engineeiing Experiment Station.

during the tests. All the tests were made with the steam pressure at or above atmospheric pressure.

The temperatures of the water were measured by mercury thermometers graduated in tenths of a degree, and readings could be estimated to hundredths of a degree Cent. The

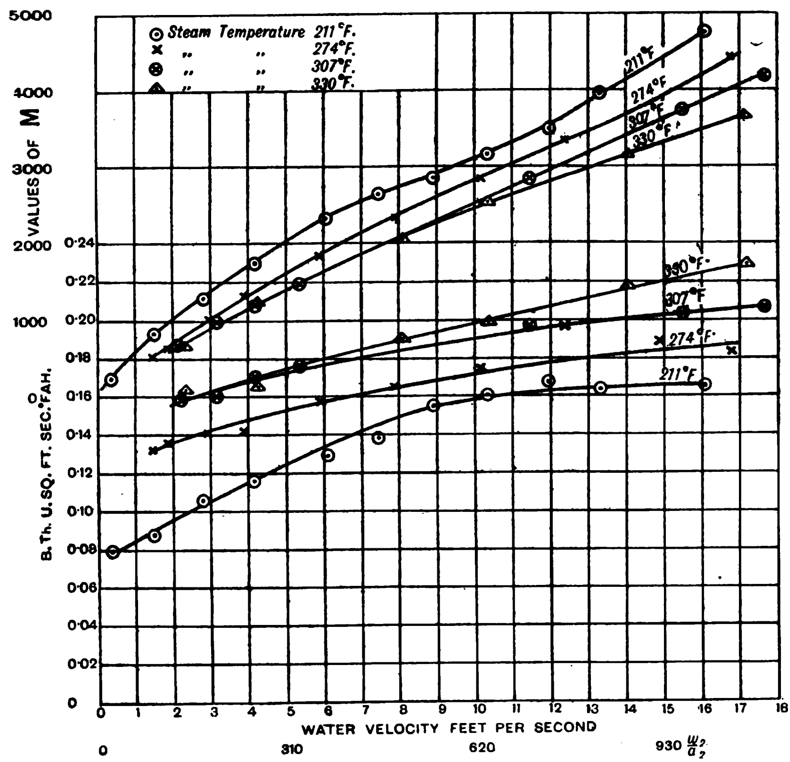


FIG. 59.—Rate of heat transmission from Clement and Garland's condenser experiments.

steam temperature was also taken by mercury thermometers with the bulbs in direct contact with the steam, allowance being made for the error due to the pressure on the bulb. The temperature of the steam wall of the tube was obtained by means of copper-constantan thermo-couples and a Siemens and Halske millivoltmeter. The couples were placed at G in Fig. 58. The ends of the copper and the constantan

wires forming the couple were soldered into small holes, about $\frac{1}{32}$ in. diameter and about $\frac{1}{64}$ in. deep, which were drilled in the surface of the tube. The leads were brought out through small glass tubing to the flanges at HH, and then between two rubber joints placed between the flanges. The mean temperature of the tube was taken as the average of the temperatures taken with the two couples.

The inlet temperature of the water varied between different tests only from 57° F. to about 67° F., and four sets of tests were made with steam temperatures varying from about 211° F. to about 330° F.

The rates of heat transmission from the steam to the water have been calculated from the tabulated values in the bulletin.

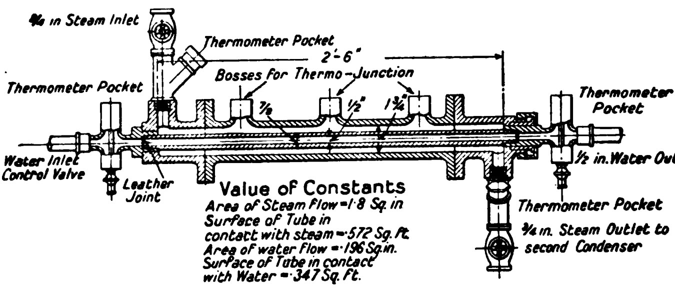


FIG. 60.—Condenser used by G. C. Webster.

The results are plotted in Fig. 59 on a water velocity base for the tests made at the various steam temperatures, and the curves suggest that the higher the steam temperature the greater is the rate of heat transmission. This may be due to the increased density of the steam as distinct from the temperature, and possibly to somewhat higher steam velocities. As regards the tube temperatures it is proposed to discuss this later on p. 196.

Using equation 5, p. 29, the various values of M were calculated and also plotted in Fig. 59. The steam temperatures T_s were taken to be those recorded in the tests.

The condensation of steam at pressures above the atmospheric pressure has also been investigated by G. C. Webster.*

* "Some Experiments on the Condensation of Steam," *Trans. Inst. Engs. and Shipbds. in Scotland*, Vol. LVII, 1913–14.

These experiments were made in the mechanical engineering laboratories at The Royal Technical College, Glasgow, with the apparatus shown in Fig. 60. This consisted of a single copper tube condenser, with the steam condensing on the outside and the water flowing inside the tube, the copper tube being adopted to ensure uniformity of the material of the tube. Special arrangements were made at the end connections to the tube to minimise endwise conduction of heat along the tube, and all these end connections were well covered to prevent excessive losses of heat to the atmosphere. The thermometer pockets were also specially designed to minimise the influence of conduction along the pocket. The temperature of the tube was obtained at two or three positions by means of thermo-junctions arranged as shown in the illustration, the details of which are fully discussed on p. 122 in *The Measurement of Steady and Fluctuating Temperatures.* When running the experiments the steam was led to a second surface condenser to complete its condensation, so that the weight of steam passing through the condenser could be measured in a tank on a weighing machine.

A series of tests with constant water velocity and constant inlet temperature, the tube temperatures being varied, was run for each of the following velocities : 1·75, 2·74, 3·92, 6·46, 9·8 and 12·15 ft. per sec. Also, for pressures of 18, 26, 32, 37, 47, 70, and 90 lb. per sq. in. abs. a series of four experiments with about constant steam velocity and with various surface temperatures was run, this series being repeated for four or more different steam velocities.

The cooling water was taken from the town's main, entering the tube at a temperature of about 45° F. It was found that the flow could be maintained constant within a variation of 1% or less, and the quantity was measured in tanks on weighing machines sensitive to within 2 oz. The thermometers used were calibrated against standard thermometers. The steam supply was obtained from a steam main and entered the condenser under approximately dry saturated conditions. The temperature of the steam leaving the condenser was usually not more than 2 or 3 degrees Fahr. below the inlet temperature. This drop would probably be partly due to

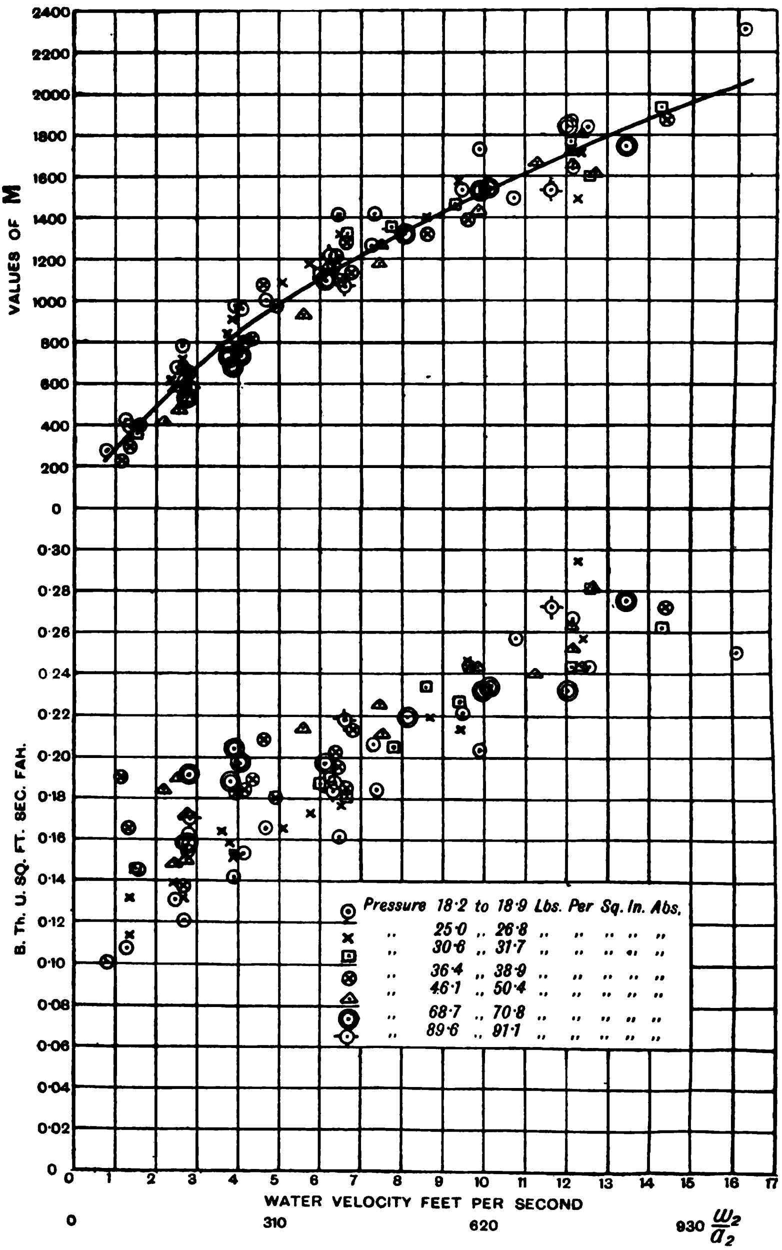


FIG. 61.—Rate of heat transmission from Webster's condenser experiments.

a small drop of pressure through the condenser and partly to the increased partial air pressure as the steam condensed, though it is evident that the amount of air in the steam could only be quite small, in fact, only the air in solution in the boiler feed water.

For these experiments the writer has calculated the rates of heat transmission from the steam to the water by means of equation 2, p. 121, taking as T_s the inlet steam temperature. The values obtained are shown plotted in Fig. 61 on the base of water velocity. An inspection of the points indicates that the higher the steam pressure and temperature the greater

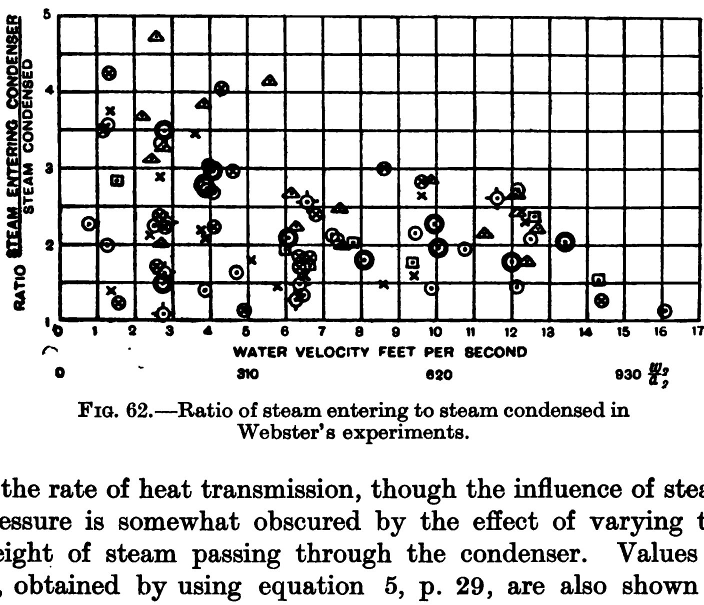


FIG. 62.—Ratio of steam entering to steam condensed in Webster's experiments.

is the rate of heat transmission, though the influence of steam pressure is somewhat obscured by the effect of varying the weight of steam passing through the condenser. Values of M, obtained by using equation 5, p. 29, are also shown in Fig. 61, again using the inlet temperature of the steam T_s. These values, though erratic in some cases, give a fairly definite indication of the variation of M with the water velocity. For the purpose of reference the ratios of the amount of steam entering the condenser to that condensed have been plotted in Fig. 62 on the water velocity base.

That the weight of steam flowing through the condenser had a small influence on the rate of heat transmission in these experiments is evident from the results shown in Fig. 63.

Here the rate of heat transmission from the steam to the water has been plotted on the base of pounds of steam entering the condenser per minute, the numbers opposite the points signifying the velocity of the water in feet per second. Points having similar water velocities have been joined by lines, and,

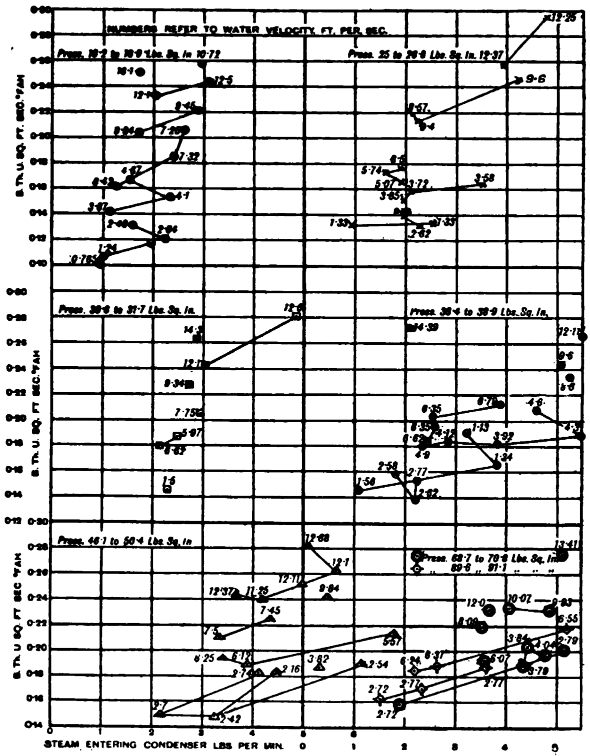


FIG. 63.—Rates of heat transmission for different series of tests from Webster's condenser experiments.

generally speaking, indicate a small increase in the rate of heat transmission when the weight of steam-flow increases, which seems to be the most marked at high water velocities, as would also be anticipated to some extent from the results of the writer's experiments shown in Fig. 55, p. 133. The question of the tube temperatures in these experiments is deferred to p. 193.

Comparatively few results are available of any extensive series of experiments made on multiple-tube surface condensers, or, at any rate, where there is also evidence of reasonable precautions having been taken to ensure accurate measurements; and it may be further remarked that, although the experiments available were made within recent years, in some cases the condenser arrangements used are now considered to be somewhat out of date.

An elaborate series of tests have been conducted by R. W. Allen,* on a surface condenser of 300 sq. ft. tube surface (steam side) where the lower rows of tubes, representing about 10% of the total tube surface, were submerged in the water of condensation, the effective steam condensing surface being therefore about 270 sq. ft. The following values represent the principal dimensions of the tubes :—

Number of brass tubes	464
Length between tube plates	4 ft.
External diameter of tubes	·625 in.
Internal „ „ „	·529 „
Number of water passes	2

The condenser shell was cast iron and circular in section.

The steam for condensation was taken from the main steam supply of the works boilers, at a pressure of 200 lb. per sq. in., with a superheat of about 80° to 90° F., and passed through a reducing valve from which it issued at a pressure varying between 9 lb. and 26 lb. per sq. in. It was then led to the bottom of a large circular receiver, whence it passed through a short length of cast-iron pipe, 15 in. diameter, direct to the condenser. The receiver contained water to a depth of about 18 in., and arrangements were made for admitting water as required from the town mains. Inside, and resting on the bottom of the receiver below the water-line, a perforated copper pipe was fitted, and the steam from the reducing valve entered this pipe and passed through the perforations into the water and up into the steam space. The steam thereby lost nearly all its superheat and gave practically dry saturated

* "Surface Condensing Plants and the Value of the Vacuum Produced," *Proc. Inst. C. E.*, Vol. CLXI, 1904–05, Part III.

steam at the condenser pressure. In the upper portion of the receiver a perforated diaphragm was fitted to prevent moisture passing over into the condenser with the steam. Special precautions were taken to prevent leakage of air into the receiver from the atmosphere.

The air-pump was of the Edwards type, three throw, the diameter of each barrel being 10 in. and the stroke 7 in., but only one of the barrels was used during the recorded tests. The water of condensation was pumped into measuring tanks. The circulating water was obtained from a tank close to the condenser, and was cooled after leaving the condenser by means of a vertical cooling tower. The circulating pump was of the centrifugal type, direct driven by an electric motor. The quantity of circulating water was measured by a weir having a rectangular notch 15 in. wide attached to the discharge side of the condenser, and the water then flowed into the tank below the cooling tower, an auxiliary pump being used for the cooling tower. The thermometers were repeatedly checked by a standard thermometer and all the vacuum readings were recorded by mercury columns. Reference to steam tables showed that the recorded steam temperatures and vacua were in fairly close agreement.

Great difficulty was at first experienced in keeping the system air-tight on account of the numerous joints. This necessitated continual testing, and before any reading was accepted a minimum vacuum efficiency of 98·5% was maintained. Towards the end of the tests, however, a vacuum efficiency of 99% was recorded, and on many occasions it was as high as 99·5%. The term " vacuum efficiency " was taken to represent the following ratio :—

$$\text{Vacuum efficiency} = \frac{\text{Actual vacuum (barometer 30 in.)}}{\begin{array}{l}\text{Vacuum corresponding with temp. of} \\ \text{air-pump discharge (barometer 30 in.)}\end{array}}$$

Four complete sets of experiments were made with this condenser, and the quantity of steam condensed in a complete test varied approximately as follows :—

(*a*) 1500 lb. per hour, or 5·00 lb. of steam per hour per sq. ft. of tube surface (300 sq. ft.).

(*b*) 2000 lb. per hour, or 6·68 lb. of steam per hour per sq. ft. of tube surface.

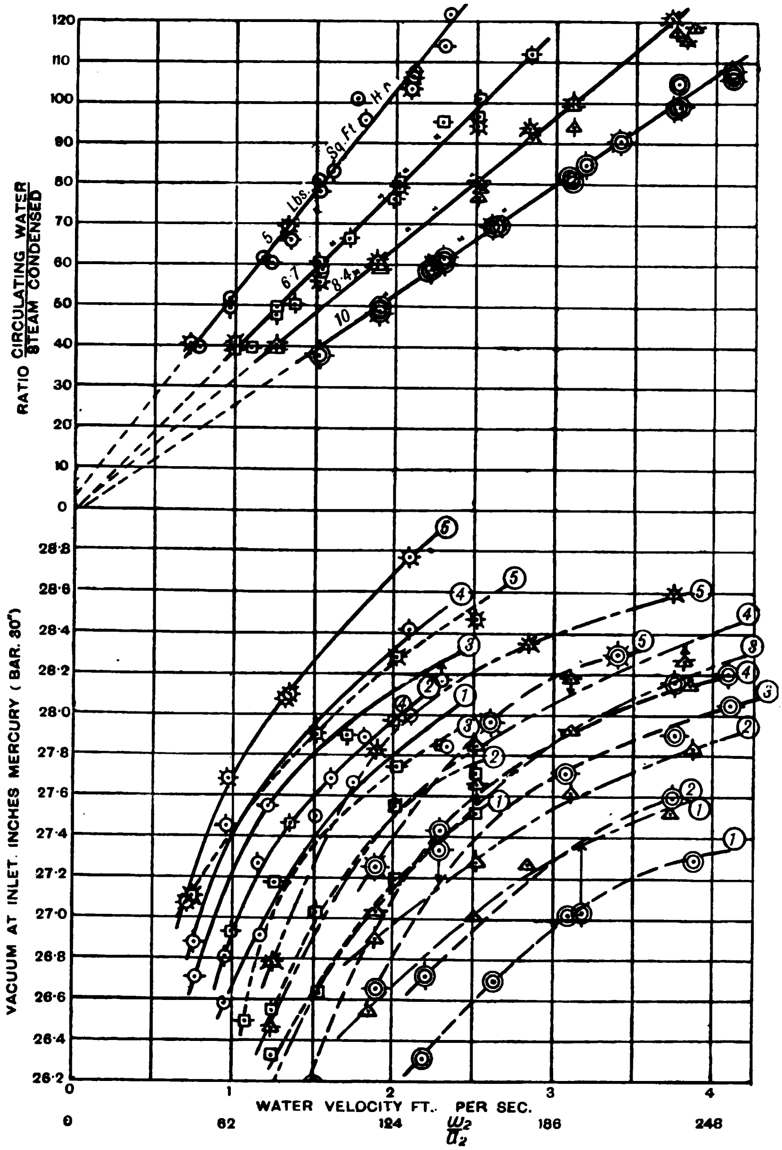


FIG. 64.—Vacuum and ratio of circulating water to steam condensed, from Allen's condenser experiments.

(*c*) 2500 lb. per hour, or 8·35 lb. of steam per hour per sq. ft. of tube surface.
(*d*) 3000 lb. per hour, or 10·00 lb. of steam per hour per sq. ft. of tube surface.

In each set of readings, the temperature of the circulating water at the condenser inlet was kept approximately constant, the values for the respective sets of tests being approximately 65° F., 70° F., 75° F., 80° F. and 85° F. In order that the readings should be reliable the tests were repeated until consistent readings were recorded. The volumetric capacity of the air-pump was the same in each set of tests, namely, about 0·75 cu. ft. per pound of steam condensed, the speed of the air-pump being varied to suit these conditions.

The recorded vacua are shown plotted in Fig. 64 on a water velocity base, from which an inspection will show how the vacuum varied with the velocity of the water, with the inlet temperature of the water, and with the rate of condensation. For purposes of comparison the ratio of the circulating water to the steam condensed has been plotted also in Fig. 64, and, evidently, mean straight lines nearly passing through the zero point represent the general relation.

The writer has calculated the rate of heat transmission, h, between the steam and the water, using equation 2, p. 121, and taking the measured steam inlet temperature as T_s. The values obtained, based upon 270 sq. ft. of effective surface, are shown in Fig. 65 on the base of water velocity. Although the values seem to be somewhat erratic the results show that in general the value of h increases with the increase in the temperature of the inlet circulating water. The reason for this is not at first evident, but it is probably mostly due to the influence of the conditions on the steam side of the tubes, that is, of the relative amounts of steam and air present at the various parts of the condenser. To some extent also it is due to the reduced viscosity of the water, as represented by equation 11, p. 119, of *Heat Transmission by Radiation, Conduction, and Convection*. It might have been expected that an increase in the rate of condensation would cause an increase in the value of h, but the results in Fig. 65 are too erratic to lead to any definite conclusion in this respect.

The values of M from equation 5, p. 29, have been calculated and are shown plotted in Fig. 65.

Extensive series of experiments on surface condensers have been made by Professor R. L. Weighton* at the Armstrong

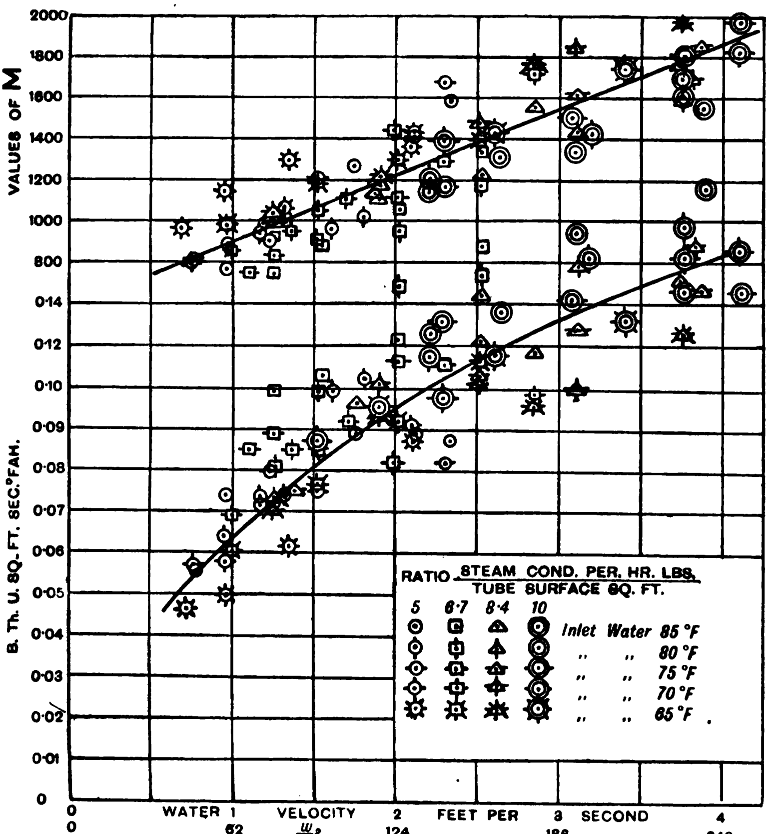
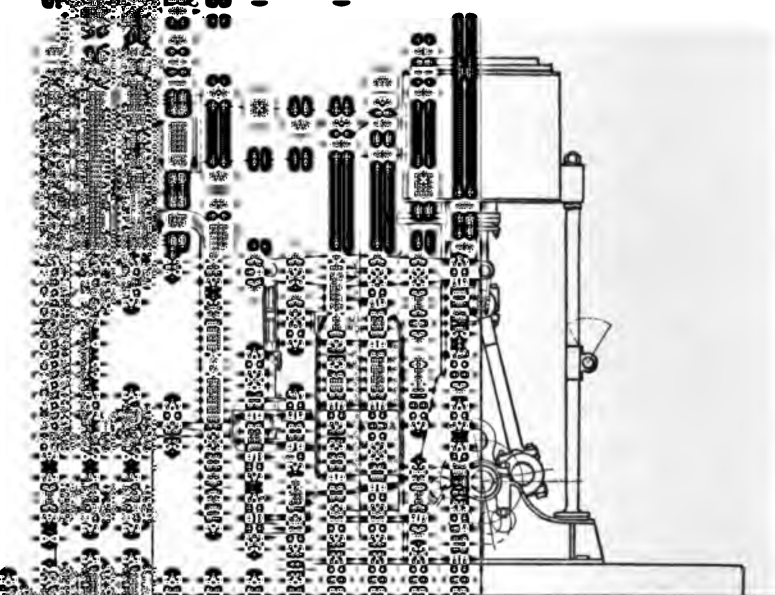


FIG. 65.—Rate of heat transmission from Allen's condenser experiments.

College, Newcastle-on-Tyne. Four different condensers were used connected up to the low-pressure cylinder of a quadruple expansion engine. These condensers were of two types, referred to subsequently as "Old Type" and "New Type." In Fig. 66 is shown a transverse section through the old con-

* "The Efficiency of Surface Condensers," *Inst. of Naval Arch.*, 1906.

[illegible] rs. The old condenser [illegible] imental engines in the [illegible] angular in cross-section, [illegible] l within recent years for [illegible] the figure. The total [illegible] was 170 sq. ft. The [illegible] 4 ft. long between the [illegible]



[illegible] ssor Weighton's experiments.

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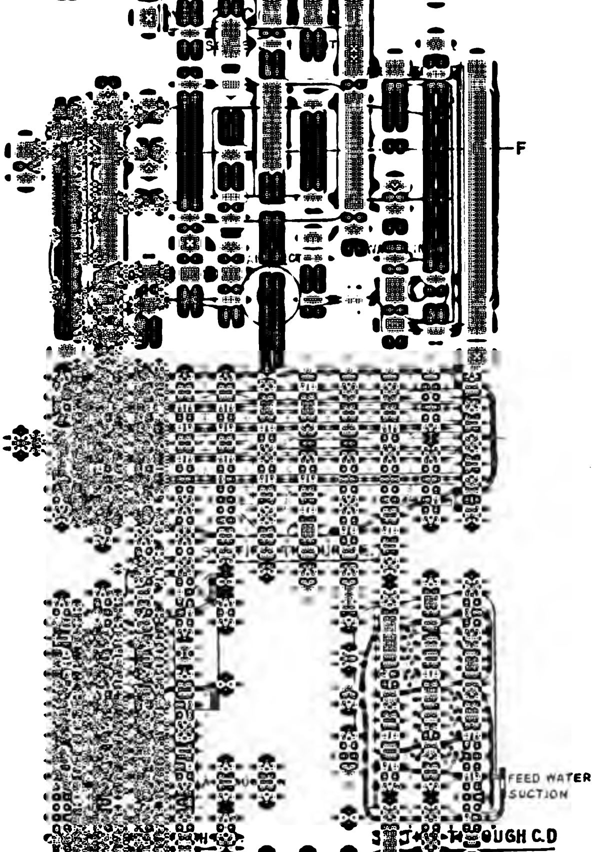
[illegible] contained 170 sq. ft. of [illegible] l diameter by 4 ft. long, [illegible] condensing water circu- [illegible] ve tube length of $4 \times 5 =$ [illegible] too large for even the [illegible] ngine could take, and [illegible], containing 100 sq. ft.

[illegible] ameter by 4 ft. [illegible] ade to circulate [illegible] 32 ft.



[illegible] ed in [illegible] experiment.

[illegible] ors and a circula- [illegible] ing an effective [illegible] hown in Fig. 67, [illegible] generally. No. 2 [illegible] the amount of [illegible] with tubes only

2 ft. 6 in. long, giving a surface of 62 sq. ft., but otherwise the same as No. 2 condenser, the water circulating four times.

One feature of these new condensers was the compartment drainage of the water of condensation. It is seen that these condensers were divided into three compartments by two diaphragms, somewhat inclined to the horizontal, and it was intended that the water of condensation in each of the three compartments should drain off directly from that compartment, the main idea being to prevent the water from the tubes in one compartment dripping over the tubes below, and to drain the water at the highest possible temperature. It will be noted also that the area available for the flow of steam decreased from compartment to compartment. Care was taken that in all cases the tubes were in a clean state when the tests began, so that, in this respect, the different types might be on the same basis.

A large number of tests were made, both on the old and on the new condensers, using the air-pump attached to the condenser of the experimental engines. This pump is of the usual vertical single-acting bucket type, driven by levers from the L.P. crosshead, and is $8\frac{1}{2}$ in. diameter by 9 in. stroke. Throughout these tests the engines were run at a standard speed of 140 revolutions per minute, thus fixing the pump capacity per unit of time.

Subsequently a three-throw Edwards air-pump was installed and driven independently by electric motor. These pumps were purposely made of somewhat abnormal size and the motor was capable of a wide range of speed. Each pump was 8-in. diameter by 8-in. stroke, and so arranged as to be capable of working single, double, or triple, in series or in parallel, and with or without a cold water spray in the base of the air-pump suction pipe.

The engine exhausting into these condensers was quadruple expansion, the cylinder diameters being 7, $10\frac{1}{2}$, $15\frac{1}{2}$, and 23 in. by 18-in. stroke. The steam supplied to this engine had a maximum pressure of 210 lb. per sq. in., and was superheated to an average of about 50° F., measured at the high pressure steam chest.

After making a large number of preliminary experiments to ascertain the best method of experimenting the following method was adopted. For a given condenser the steam supply to the engine was adjusted to give the desired rate of condensation per square foot of tube surface. Then a series of tests—usually five—were made at this rate with different quantities of condensing water. Care was taken that the intervals between the tests were of sufficient duration to enable the new conditions to attain stability. The condensing water was taken directly from the city mains at an inlet temperature ranging from about 40° F. to 65° F. over the period of the whole tests. In the results about to be considered, however, the water inlet temperature was practically constant over each set of the series of tests, and then varying only between 40° F. and 50° F. in the different sets of tests.

Several series of tests were made with a falling vacuum as against a rising vacuum and it was found that, provided a sufficient interval elapsed between any two consecutive tests, there was no sensible difference in result as between falling and rising vacuum conditions. Precautions were taken against excessive air leakage into the condenser at the piston-rod glands and exhaust pipe joints, etc.

Two calibrated vacuum gauges as well as a mercury column were used on each condenser. Care was taken to measure the true vacuum by fitting a curved projecting nozzle to the gauge pipe inside the condenser and turning it round in all directions to observe the effect while a test was in progress. In no case was it found that the flow of steam past the tube or nozzle produced any "induced" effect on the recorded vacuum. Except in a few cases the measured vacuum at the bottom of the condenser was much the same as that measured at the top, and since it is the vacuum at the top of the condenser which influences the steam engine, in the subsequent discussion this has been taken to represent the condenser vacuum.

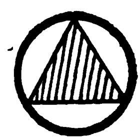


FIG. 68.—Core of wood in tubes in some of Weighton's condenser experiments.

In some of the series of tests cores of wood of triangular section, shown in Fig. 68 in full-size section, were

inserted in the tubes in order to increase the ratio $\frac{l}{m_2} = \frac{\text{effective tube length}}{\text{hydraulic mean depth of tube}}$. These cores were merely laths of hard wood rough from the saw, about 2 in. longer than the tubes, and simply inserted in the tubes without any fastening whatever.

As already mentioned on p. 152, tests were made by Professor Weighton with various air-pump arrangements, and a typical example of the results obtained is given in Fig. 69 for the No. 2 condenser, the conditions obtaining

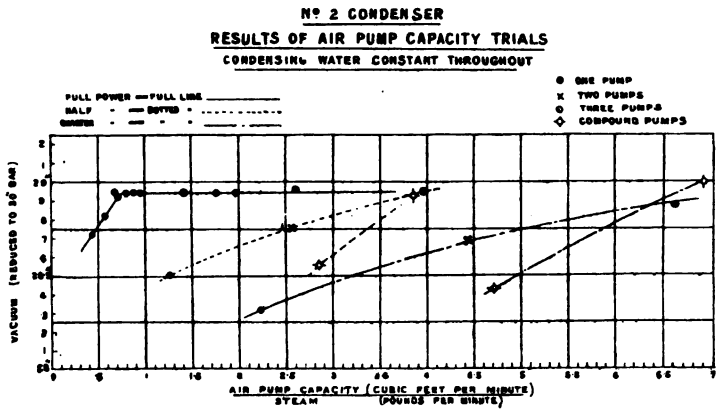


FIG. 69.—Influence of air-pump capacity on vacuum, Weighton's experiments.

being indicated in the figure. Three different sets of curves are shown, viz.—for full power, half power, and quarter power, developed by the engines. It must be remembered that the air leakage is likely to increase in quantity with a reduction of load on the engines on account of the admission pressures in the L.P. cylinder being below atmospheric pressure at low loads. Thus the ratio of air to steam entering the condenser increases rapidly as the load decreases for the reason that the quantity of air leaking probably increases and the quantity of steam used decreases. The results in Fig. 69 show that, at full load, increasing the air-pump capacity beyond that giving ·7 cu. ft. per pound of steam condensed did not result in any appreciable increase of

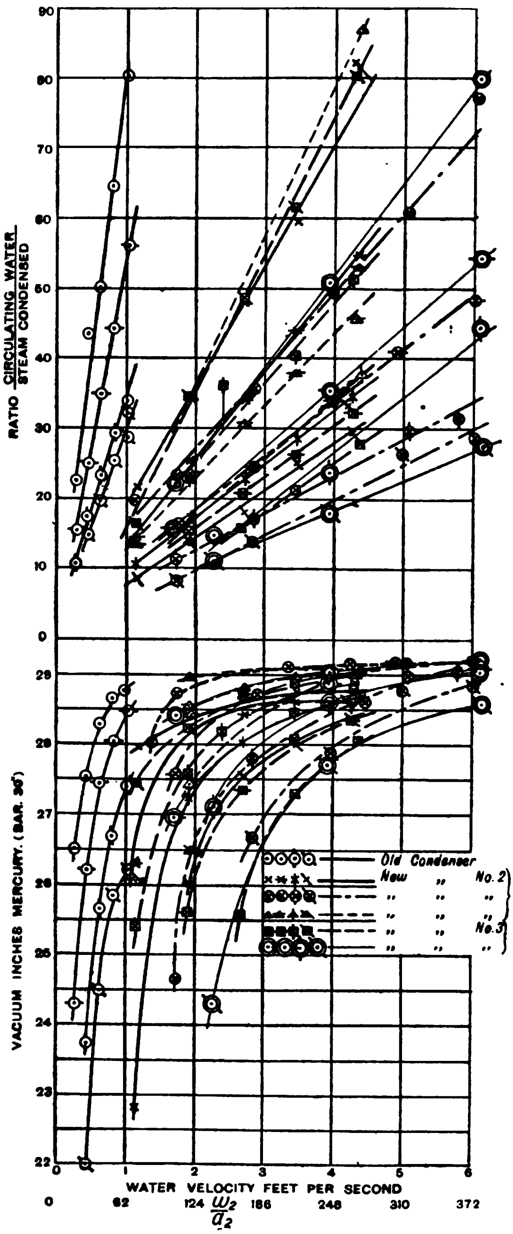


FIG. 70.—Vacuum and ratio of circulating water to steam condensed, from Weighton's condenser experiments.

vacuum. At half and quarter loads, however, an increase of air-pump capacity caused a definite increase of vacuum throughout the range of capacity employed.

For purposes of reference and comparison, the recorded vacua (barometer 30 in.) obtained in the various tests with the several condensers are shown plotted* in Fig. 70 on a base of water velocity through the tubes. The curves show distinctly that in each condenser the vacuum decreases for any given water velocity as the rate of condensation increases, which, of course, is what would be expected when the inlet temperature of the water is practically constant. There is also shown in Fig. 70 the ratio between the condensing or circulating water and the $\frac{\text{Heat to water, B.Th.U.}}{1000}$. The latter mentioned value has been taken in preference to the weight of steam condensed, and it assumes that a pound of steam *in* condensing would give about 1000 B.Th.U. to the water. The general relation between the above-mentioned ratio and the water velocity is seen to be a series of straight lines nearly passing through the origin, as was the case in Allen's experiments shown in Fig. 64, p. 147.

The writer has calculated from the tabulated results of Weighton's experiments the rates of heat transmission, h, between the steam and the water, using equation 2, p. 121, and taking the steam temperature T_s to be the measured inlet temperature. The results are shown plotted in Fig. 71 on a water velocity base. An examination of the points in this figure shows the very large variations in the rate of heat transmission obtained at any given water velocity, and without doubt this is principally due to the influence of air in the condenser. It will be noted that, generally speaking, the higher the rate of condensation the higher is the rate of heat transmission shown by the points in this figure. Making use of equation 5, p. 29, the corresponding values of M are shown in Fig. 71, and these indicate similar wide variations in their values.

Professor E. Josse made some experiments on the surface

* Refer to Fig. 71 for the description of the character and meaning of the different points.

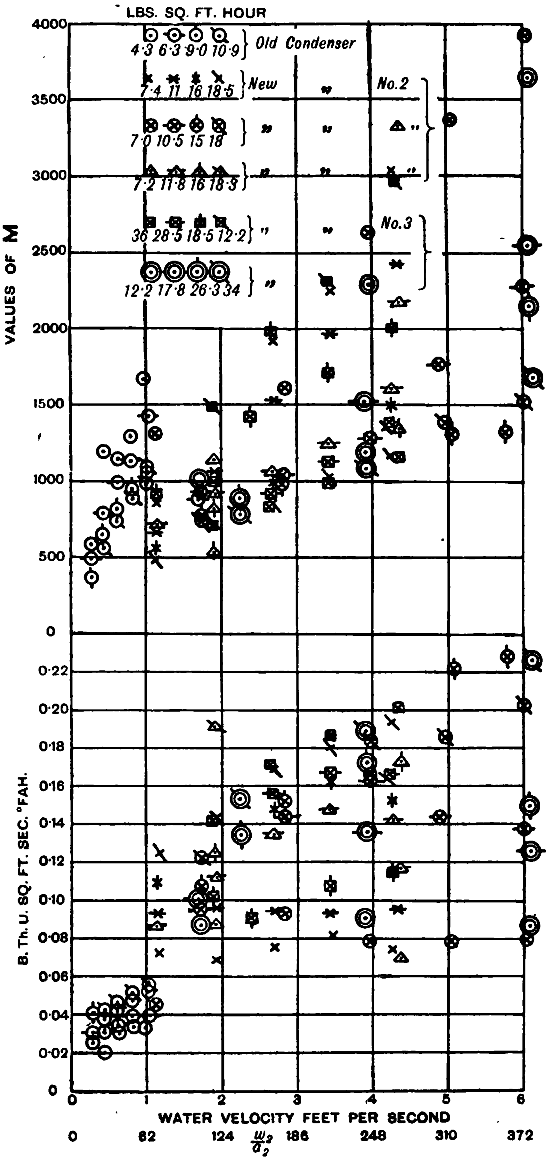
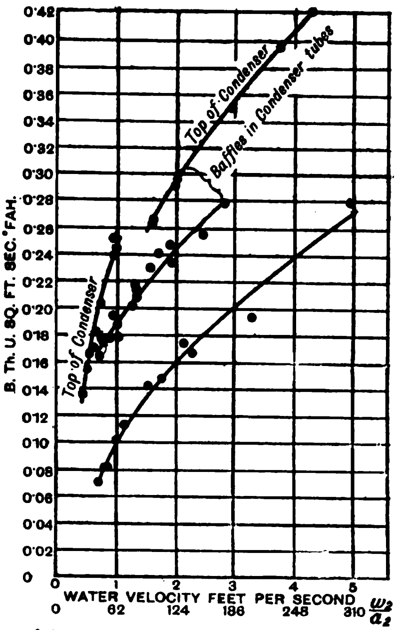


FIG. 71.—Rate of heat transmission from Weighton's condenser experiments.

condenser referred to on p. 163. The rates of heat transmission obtained are shown in Fig. 72 plotted on the base of water velocity, for the usual vacua varying from about 96% down to about 89% of the normal barometric pressure. In one set of tests baffle strips or "retarders," supplied by Pape, Henneberg and Co., of Hamburg, were inserted in the tubes. Compared with the results without baffles in the tubes it is seen that in this case there was apparently a large increase in the rate of heat transmission by their use.

Having regard to the increase of resistance to the flow of the water it is doubtful whether retarders in condenser tubes are really more effective than plain tubes. In other words, the same pressure difference with plain tubes of suitable length could give a higher velocity of flow and probably as high a rate of heat transmission as when retarders are used suited for the same total resistance to the flow.



fFig. 72.—Rate of heat transmission rom Josse's condenser experiments.

The Influence of Air in the Steam on the Rate of Heat Transmission.—It was clearly demonstrated on p. 110 that most of the air which finds its way into a surface condenser under ordinary conditions of operation is due to leakage, and that only a small proportion is due to the air originally in solution in the feed water supplied to the boilers. The air affects the transmission of heat in two ways :—

1. As was shown on pp. 101 to 104 the pressure of the air reduces the partial pressure and the temperature of the steam for a given vacuum as the mixture of steam and air flows through the condenser ; and

[illegible] act between the [illegible] if the condenser [illegible] the steam side

[illegible] ich demonstrate [illegible] tion and on the [illegible]

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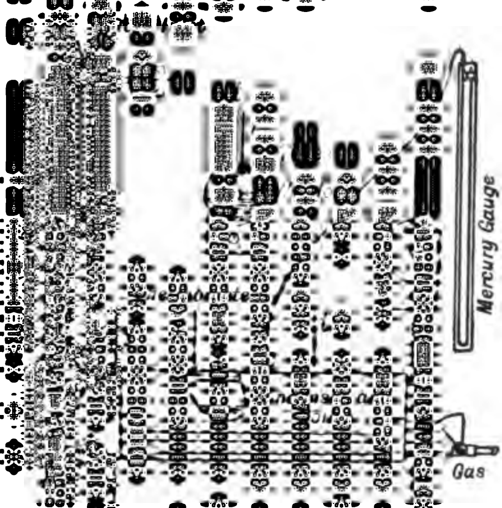

[illegible] by [illegible]

[illegible] n a small surface [illegible] s of the experi- [illegible] rface condensers, [illegible] having stagnant [illegible] der about 3 ft. [illegible] in Fig. 73. All [illegible] arred to prevent [illegible] tion of $\frac{1}{50}$ in. of [illegible]. Two tubes, of [illegible] series, having a

[illegible] Inst. of Engs., 6th [illegible] XXI.

d an external surface of
m in the cylinder. The
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Precautions were taken
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e course of each experi-
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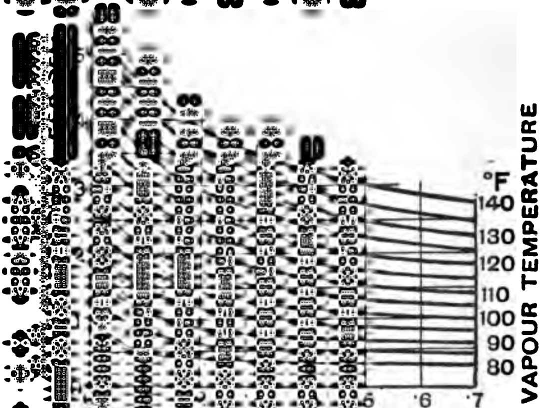
air-pump or syringe to
e heating gas-jets were



h condenser tube at
ith's experiments.

he mercury gauge or the
about 2 lbs. per sq. in.
m was then allowed to
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he observation window.
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ies of air could then be
a special connection and
ssure was used to check

During an experiment cooling water was allowed to flow through the tube at a definite rate, and the steam condensed on the outer surface, the temperature conditions being kept constant by regulation of the gas-jets. Thus the water of condensation would fall back into the boiling water. Tests were first made at steam temperatures from 70° F. to 140° F., at 10° intervals, but without air in the vessel. Then a constant steam temperature was maintained and the air pressure varied between tests, the process being repeated at various tempera-

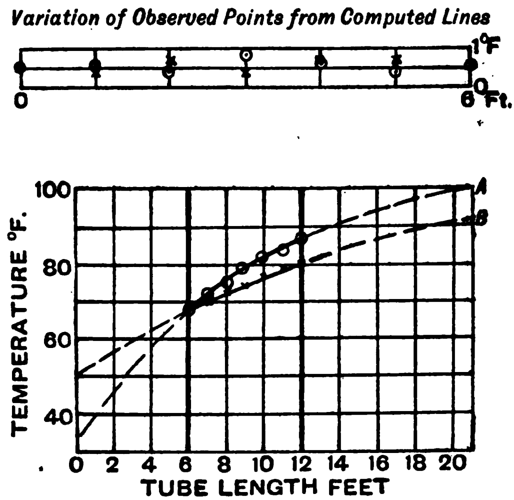


FIG. 75.—Temperature of water along condenser tube in Smith's experiments.

tures. In all these tests the cooling water was arranged to flow at the constant rate of 10 lb. per min., that is, about 1·49 ft. per sec.

In the paper the results obtained were plotted and a series of curves drawn through the points, but as the points lay on the curves in nearly every case it would hardly be worth while reproducing them here. The results are sufficiently well represented by the interpolation curves shown in Fig. 74. The base line represents the partial air pressures reduced to 70° F., the right-hand scale shows the vapour temperatures and the left-hand scale the B.Th.U. transmitted per sq. ft.

per second. The reduction of the air pressure to 70° F. was obtained as follows: If $x°$ F. is the temperature in the vessel during the experiment, then,

$$\frac{\text{Partial air pressure at } 70° \text{ F.}}{\text{Partial air pressure at } x° \text{ F.}} = \frac{460+70}{460+x}.$$

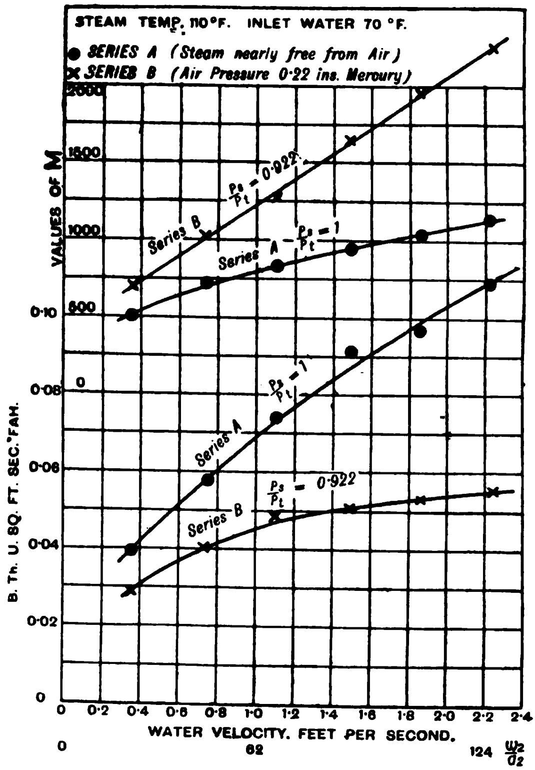


Fig. 76.—Rate of heat transmission from Smith's condenser experiments.

Referring to Fig. 74 it would be seen that at small values of the air pressure the heat transmitted decreases rapidly with increase of air pressure. It must be again noted, however, that these results refer only to what might be called stagnant conditions, since the steam side of the tube was not swept by a stream of fluid.

In Fig. 75 the temperatures of the water along the 6 ft.

length of tube are shown plotted for two tests. The points were found to fall in very closely with the law that the transmission of heat is proportional to the difference of temperature between the steam and water. The mean lines have been extended both ways according to this law to show more clearly their general nature, line A referring to nearly air-free steam and line B to steam with a partial air pressure of ·22 in. of mercury referred to 70° F. The vapour temperature in both cases was 110° F. and the velocity of the cooling water 1·49 ft. per second. The curves in Fig. 76 show the corresponding rates of heat transmission from the steam to the water for the two conditions just specified in connection with Fig. 75, calculated by means of equation 2, p. 121.

The values of M in the equation 5, p. 29, are also shown plotted in Fig. 76 for the conditions just specified.

Professor E. Josse,* of Charlottenburg, has made some experiments on a surface condenser in connection with a Parsons turbine of 300 kilowatts. The following are the principal dimensions of the condenser :—

Cooling surface		89 sq. m. (956 sq. ft.)
Tubes	Internal diameter	18 mm. (·71 in.)
	External diameter	20 mm. (·79 in.)
	Length between tube plates	2300 mm. (90·5 in.)
Number of tubes (upper set)		346
„ „ „ (lower set)		342
	Total	688

Total area cross-section of tubes (upper set) ·0879 sq. m. (136 sq. in.)

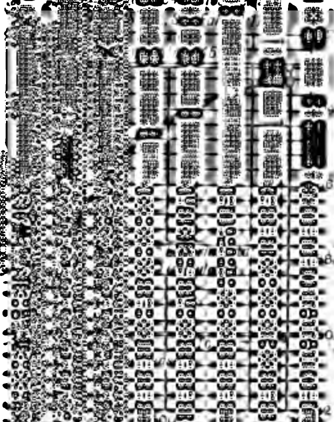
„ „ „ „ „ (lower set) ·0868 sq. m. (135 sq. in.).

Thermo-couples were inserted in the water in the tubes to measure the temperature of the water at different points along the length. The black dots in Fig. 77 show the experimental points obtained with very little air in the condenser, and the circles indicate calculated points obtained by taking

* "Surface Condensers for Steam Turbines," *Engineering*, Dec. 11th, 1908, being a summary of paper read at summer meeting of Schiffbautechnische Gesellschaft in Berlin, 1908.

...he difference of tempera- ... water. It is seen that ... the mean curve drawn ...

... the temperature of the ... urve 5 was obtained with ... exhaust steam, whereas ...6 kgm. (21·2 lb.) of air ... oth cases there was little



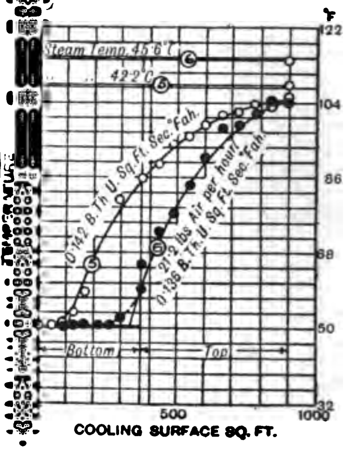


FIG. 78.—Average temperature of water along length of condenser tubes in Josse's experiments with air leakage into condenser.

... f the water for some dis- ..., and curve 6 shows that ... occurred at all in the ... might be compared with ...

... d that, with this 300-kw. ... ry conditions, the normal ...5 kgm. per hour (·55 lb.) ... were condensed per hour. ... mance of condensers has ... F. C. McBride, of Phila-

delphia.* He gave the results of a few experiments on a large condenser attached to a steam turbine and in which the leakage of air was fairly large. The condenser evidently had three water passes and the approximate temperatures of the steam were measured between the passes as well as those of the circulating water. The condenser dimensions were not given, but the data given in Tables 12 and 13 are based on some of the results given in the paper.

TABLE 12

Test Number.	1	2	3	4
1. Ratio steam being condensed to guaranteed capacity of condenser (water inlet 70° F. and 28 in. vacuum)	·74	·67	·38	·275
2. Vacuum in exhaust (30 in. bar.) in. mercury	28·03	28·18	28·28	27·34
3. Corresponding temperature, °F.	101·0	97·8	95·9	109·4
4. Air pump suction temperature, °F.	56·7	58	58	56·2
5. Equivalent steam pressure, in. mercury	·46	·48	·48	·45
6. $\frac{\text{Air pump displacement}}{\text{Volume water of condensation}}$	52·6	58·0	105·0	143·5
7. Air-pump displacement per lb. water of condensation, cu. ft., approximate	·88	·97	1·75	2·4
8. Steam condensed per sq. ft. per hour, lbs.	5·62	5·1	2·85	2·07

TABLE 13

1	2	3	4	5	6	7	8	9	10	11
Test No.	Nest of tubes.	Steam temperatures, °F. (Approximate.)			Water temperatures, °F.			Temp. diff. °F.	B.Th.U. per sec. per sq. ft.	B.Th.U. per sec. per sq. ft. per °F. diff.
		Entering.	Leaving.	Mean.	Entering.	Leaving.	Mean.			
1.	Top	101·0	91·1	96·0	56	66	61	35	3·62	·103
	Middle	91·1	63·6	77·3	53	56	54·5	22·8	1·08	·0475
	Bottom	63·6	55·3	59·4	52	53	52·5	6·9	·36	·0525
2.	Top	97·8	91·7	94·8	56	66	61	33·8	3·26	·0966
	Middle	91·7	62·0	76·8	53	56	54·5	22·3	·97	·0434
	Bottom	62·0	56·0	59·0	52	53	52·5	6·5	·32	·0500
3.	Top	96·0	89·3	92·6	54	60	57	35·6	1·84	·0517
	Middle	89·3	59·4	74·3	53	54	53·5	20·8	·31	·0150
	Bottom	59·4	55·6	57·5	51·6	53	52·3	5·2	·43	·0083
4.	Top	109·4	100·0	104·7	54	57·5	55·7	49·0	·76	·0161
	Middle	100·0	62·0	81·0	54	54	54·0	27·0	—	—
	Bottom	62·0	54·4	58·2	52	54	53·0	5·2	·45	·0086

* *Trans. Amer. Soc. Mech. Engs.*, Vol. 30, June, 1908.

In a large condenser it is somewhat difficult to obtain the true steam and water temperatures at the various passes and it is probable that the number of significant figures given in columns 10 and 11 of Table 13 are hardly justified by the experimental conditions. However, an examination of column 11 for any test shows roughly the influence of the air at the different portions of the condenser. An inspection of Table 12 shows that the air-pump displacement was practically constant, since the products of lines 7 and 8 are constant, and it might be inferred that the leakage of air into the condenser was nearly constant, but if anything increased as the load decreased, which is usual with steam turbines under ordinary conditions. Therefore, assuming the velocity of the circulating water was maintained constant, the differences in the values in column 11 of Table 13 as between the different tests exhibits the effect of the increase of the ratio of air to steam as the load on the turbine decreased, though probably the smaller the quantity of steam entering the condenser the greater is the liability to stagnation at some parts of the condenser.

The influence of air leakage on the performance of a condenser and on the temperature of the cooling or circulating water is again illustrated by the results given in Table 14. This condenser, installed in the mechanical engineering laboratory at The Royal Technical College, Glasgow, is of rectangular cross-section similar to the "old condenser" tested by Professor Weighton and discussed on p. 150. The tube condensing surface is 400 sq. ft., with two water passes. As shown in Table 14, in columns 14 and 15, the velocity of the water was very low, in fact, much below what is now considered ordinary practice, but higher velocities were not necessary in this case because the condenser is of large capacity compared with the amount of steam to be condensed. During the tests 1 to 6 the condenser was connected to an experimental impulse turbine designed to exhaust at about atmospheric pressure. The steam supply to the turbine glands was purposely shut off and thus in these tests there was a large leakage of air into the turbine casing.

An inspection of the cooling water temperatures, columns 11, 12, and 13, shows that from tests 1 to 4 the bottom or first

pass only became really effective when the condenser was supplied with comparatively large quantities of steam. In tests 4 to 6 the air-pump suction temperature was measured, and inspection of column 27 shows how low were the values of $\frac{P_s}{P_t}$, where P_s is the partial steam pressure at the air-pump suction and P_t is the total pressure at the top of the condenser. The results in column 25, calculated by use of equation 2, p. 121, illustrate how greatly the rate of heat transmission is influenced by the conditions on the steam side of the tubes.

The tests 7 to 13 were made when condensing steam from a reciprocating engine, there being not more than a small leakage of air into the condenser. It will be noticed that the rates of heat transmission, shown by columns 23, 24, and 25, and calculated from equation 2, p. 121, were much the same in the bottom pass as in the top pass. The temperatures of the cooling water given in columns 11, 12, and 13, and the corresponding transmission of heat given in columns 17, 18, and 19, show clearly how effective was the bottom pass under these conditions of operation. The air-pump suction temperatures in column 7, by consulting steam tables, gave the data for the ratio $\frac{P_s}{P_t}$ in column 27.

The low rate of heat transmission sometimes obtained in the lower portions of a condenser with the steam entering at the top, and largely due to the presence of air, is further well illustrated by the diagrams of temperatures shown in Fig. 79 and Fig. 80. These were obtained from Professor Weighton's experiments, the results in Fig. 79 referring to his No. 2 condenser described on p. 151, and those in Fig. 80 to his No. 3 condenser. Both condensers had four water passes, as is indicated at the base of the diagrams. The plotted points indicate the measured temperatures at the beginning and end of each pass, and those for each test are joined together by straight lines. The condenser vacuum and the weight of steam condensed per hour per square foot of tube surface $\frac{W}{S}$ are marked against each test in the diagrams in Fig. 79. In

TABLE 14

SURFACE CONDENSER

Diameter of tubes	Outside	·75 in.
	Inside	·654 in.
Length between tube plates		5 ft. 9 in.
Number of tubes		355.
Cooling surface of tubes (outside), sq. ft.	Bottom pass	196
	Top pass	204
	Total	400
Area for water-flow, sq. ft.	Bottom pass	·405
	Top pass	·421

Conditions of Tests.	No. of Test.	Bar. press. in. mercury.	Vacuum top of condenser, in. mercury.	Abs. press. top of condenser, in. mercury, P_t	Measured steam temperatures, °F.	Air-pump suction temperatures, °F. No. I pump.	Wet air-pump, discharge temperatures, °F.			Cooling water temperatures, °F.		
							No. I pump.	No. II pump.	Mean.	Inlet.	Middle.	Outlet.
Exhaust	1	30·26	3·67	26·59	203·5	—	95·7	95·6	95·6	48·0	50·0	140·7
steam	2	,,	3·50	26·76	204·3	—	110·2	116·8	113·5	48·0	53·0	153·
from	3	,,	3·29	26·97	205·2	—	152·8	151·0	151·9	48·0	101·4	187·7
turbine	4	,,	4·17	26·09	202·8	87·8	98·5	98·3	98·4	48·25	51·0	137·5
(Large air	5	,,	3·36	26·90	204·2	101·6	105·6	108·8	107·2	48·25	—	151·2
leakage).	6	,,	1·9	28·36	208·0	153·6	158	158	158	48·25	—	192·8
Exhaust	7	30·46	25·9	4·56	131	126·9	120·7	127·5	124·1	46·3	100	116·8
steam	8	29·85	26·5	3·35	120	113·4	111·7	114·6	113·1	46·8	90·4	106·8
from	9	,,	24·6	5·25	134·5	130·9	127·7	132·1	129·9	47·3	112·3	128·9
recipro-	10	30·24	26·6	3·64	122·7	114·9	111·9	116·5	114·2	48·8	89·1	108·2
cating	11	,,	24·85	5·39	136·4	130·8	127·0	132·7	129·9	49·2	109·2	127·7
engine.	12	30·4	26·2	4·2	127·3	121·0	116·5	122·6	119·5	42·2	98·8	116·2
	13	,,	26·45	3·95	125·4	119·0	116·6	120·2	118·4	41·8	83·6	105·2
1	2	3	4	5	6	7	8	9	10	11	12	13

TABLE 14 (*continued*)

Velocity of cooling water, ft. per sec.		Steam condensed per sq. ft. per hour. lbs.	B.Th.U. per hour per sq. ft. / 1000			Mean temperature difference, °F.			B.Th.U. per sq. ft. per sec. per °F.			Circulating water / Steam condensed.	$\frac{P_s}{P_t}$ At air-pump suction.
1st pass (bottom).	2nd pass (top).		1st pass (bottom	2nd pass (top).	Total condenser.	1st pass (bottom).	2nd pass (top).	Whole condenser.	1st pass (bottom).	2nd pass (top).	Whole condenser.		
·219	·211	4·3	·20	8·8	4·59	—	—	102	—	—	·0125	11·5	—
·358	·345	8·25	·82	15·9	8·49	—	—	94	—	—	·0251	9·8	—
·384	·370	12·7	9·42	14·7	12·1	—	—	63·6	—	—	·0529	6·8	—
·27	·26	5·05	·34	10·3	5·42	—	—	103·5	—	—	·0145	12·0	·051
·30	·29	6·64	—	—	6·92	—	—	95·5	—	—	·0202	10·1	·075
·375	·37	13·1	—	—	12·5	—	—	61·5	—	—	·0565	6·6	·292
·279	·269	4·65	6·9	2·07	4·44	53·6	21·6	37·2	·0358	·0266	·0331	13·5	·91
·254	·244	3·52	5·1	1·83	3·43	48·3	20·3	35·0	·0293	·025	·0272	16·25	·855
·167	·161	3·35	5·01	1·23	3·09	46·9	12·1	29·7	·0297	·0282	·0289	11·3	·882
·28	·27	3·85	5·20	2·37	3·76	51·1	22·7	36·4	·0283	·029	·0287	16·5	·82
·204	·197	3·86	5·64	1·67	3·62	51·4	16·2	34·1	·0305	·0286	·0295	11·9	·86
·204	·197	3·57	5·34	1·57	3·42	51·7	18·5	36·4	·0286	·0236	·0261	12·9	·843
·334	·322	4·93	6·43	3·20	4·79	60·3	29·7	44·5	·0297	·0299	·0298	15·3	·846
14	15	16	17	18	19	20	21	22	28	24	25	26	27

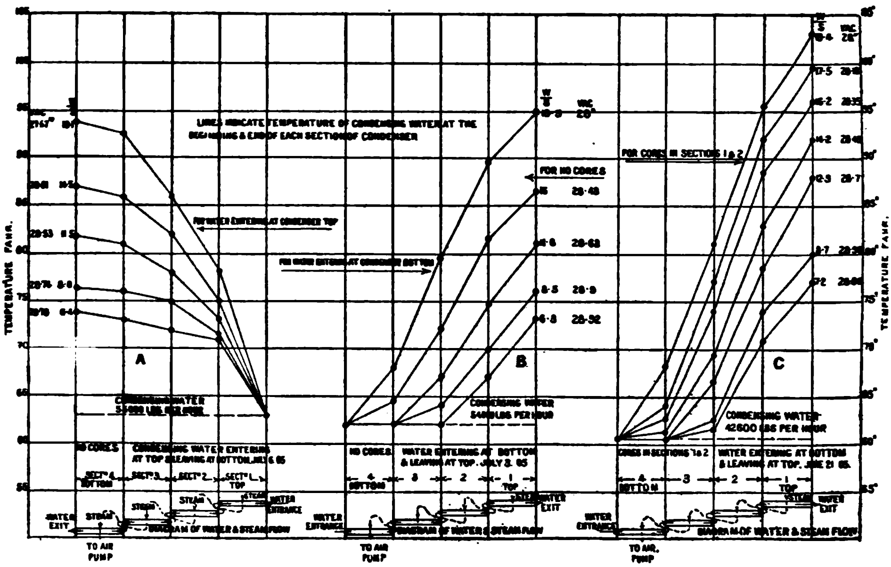


FIG. 79.—Temperatures of circulating water in Weighton's condenser experiments. No. 2 condenser.

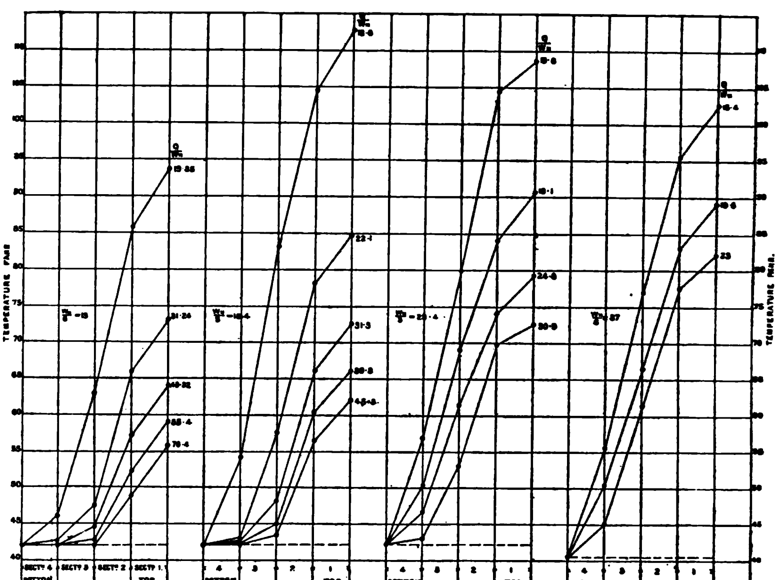


FIG. 80.—Temperatures of circulating water in Weighton's condenser experiments. No. 3 condenser.

diagram A the circulating water, 55,000 lb. per hour, entered at the top and left at the bottom of the condenser, whilst in diagram B the water, 54,000 lb. per hour, entered at the bottom and left at the top as in the usual practice. In both cases the steam entered at the top, and it would be noticed that the heat transmission in the bottom passes were very small when the values of $\frac{W}{S}$ were small, but that when $\frac{W}{S}$ was large the bottom pass was fairly effective; also, a comparison of the vacua obtained in diagrams A and B show that the usual practice of sending the water in at the bottom and out at the top of the condenser (diagram B), gave higher vacua than was obtained under the conditions for diagram A. Diagram C, Fig. 79, indicates the measured water temperatures, steam condensed per square foot $\frac{W}{S}$, and the vacua when the condenser had cores in the tubes of the two top passes of the form shown in Fig. 68, p. 153, and when passing 42,600 lb. of water per hour. This diagram C shows the same general features as diagram B.

The various series of diagrams in Fig. 80 were obtained, each with the constant value of $\frac{W_H}{S}$ marked on the diagrams, and with various values of the ratio of the circulating water to the steam condensed $\frac{Q}{W_H}$, where W_H refers to the weight of steam condensed when referred to standard heat conditions, as is explained on p. 156, and Q is the quantity of circulating water. These diagrams not only illustrate again how relatively ineffective the bottom portion of the condenser becomes when $\frac{W}{S}$ is small, but they also show clearly that the bottom is relatively more effective when $\frac{Q}{W}$ is small than when it is large.

To examine further the relations between the rate of heat transmission and the condenser conditions with respect to the presence of air the results shown in Figs. 81 to 84 have been deduced for certain of the condensers previously described.

In these figures the rates of heat transmission are plotted against the ratio $\frac{P_s}{P_t}$ at the condenser steam outlet. Fig. 81 refers to the small double-tube condenser described on p. 132 when condensing steam at atmospheric pressure, and has been derived from Fig. 55, p. 133, by selecting values from

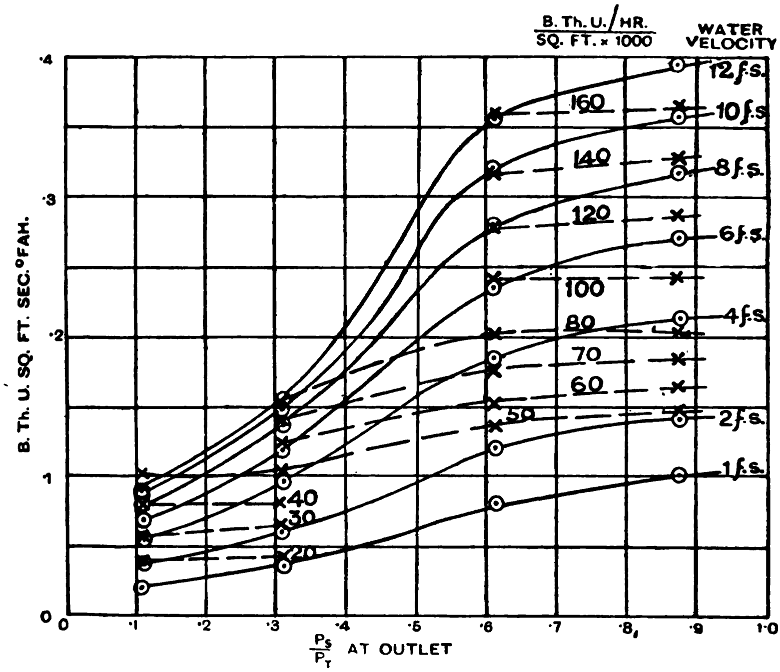


FIG. 81.—Variation of rate of heat transmission, from Royds' condenser experiments.

the curves in this figure. The full lines indicate the relation between the rate of heat transmission and $\frac{P_s}{P_t}$ at different velocities of the cooling water, and the dotted lines show how the rate of heat transmission varies with $\frac{P_s}{P_t}$ at the difference values of $\frac{\text{B.Th.U. per sq. ft. per hour}}{1000}$ given by the numbers in the figure. Also, following any constant velocity line shows

how the rate of condensation given by the dotted lines decreases with the decrease of $\frac{P_s}{P_t}$; or again, following any one of the dotted lines of constant rate of condensation it is seen how $\frac{P_s}{P_t}$ decreases with an increase of the water velocity. It is also found that when $\frac{P_s}{P_t}$ is very low an increase in the velocity of the water for any given rate of condensation has little influence on the rate of heat transmission.

The results in Fig. 82 have been obtained from the curves in Fig. 76, p. 162, and refer to the small boiler-condenser arrangement described on p. 159. The full lines again refer to constant velocities of the cooling water and the dotted lines to the constant values of $\frac{\text{B.Th.U. per sq. ft. per hour}}{1000}$ given in the figure. Although in this case the values of $\frac{P_s}{P_t}$ represent the conditions all over the steam side of the tube instead of at the outlet as in the previous case, the general relations are much the same as in Fig. 81. It will be remembered, however, that in this apparatus the steam condensed under stagnant conditions.

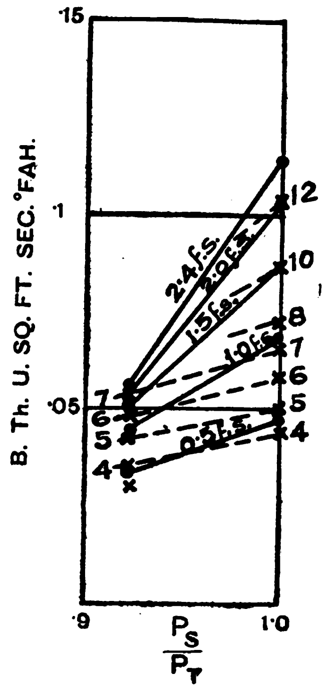


FIG. 82.—Variation of rate of heat transmission, from Smith's condenser experiments.

Considering some of the results from multiple tube condensers, Fig. 83 refers to the condenser of rectangular section described on p. 150, and tested by Professor Weighton. The full lines show the relation between the rate of heat transmission and $\frac{P_H}{P_t}$ for velocities of the water varying from ·27 to

1 ft. per sec., where P_H is the steam pressure equivalent to the hot-well temperature and P_t the pressure at the top of the condenser. The dotted lines refer to constant rates of condensation from 4·3 to 10·9 lb. per sq. ft. per hour. Whilst the general relation shown by the constant velocity lines is similar to that given by Figs. 81 and 82, the dotted lines, showing results at constant rates of condensation, slope in the opposite direction, but again are more nearly horizontal the lower the rate of condensation. In this same figure are plotted the results of the condenser tests given in

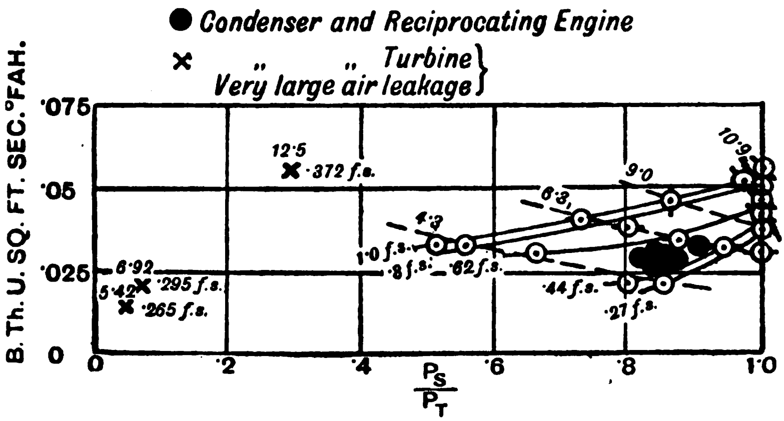


FIG. 83.—Variation of rate of heat transmission for surface condensers of rectangular section.

Table 14, p. 168, obtained on a condenser of similar form, and it is seen that the results agree fairly well.

The results shown in Fig. 84 were deduced from the results of Professor Weighton's experiments described and discussed on pp. 149–156. The lower set of curves refer to the No. 2 condenser and the upper set to the No. 3 condenser, in both cases when using ordinary air-pumps. In this figure $\frac{P_s}{P_t}$ is the ratio of the partial steam pressure at the bottom of the condenser to the total pressure. The full lines indicate how the rate of heat transmission varies with $\frac{P_s}{P_t}$ at the different velocities of the circulating water given in the figure, whilst

the dotted lines refer to the constant rates of condensation stated in pounds of steam per sq. ft. per hour. In the No. 3 condenser (upper portion of Fig. 84) the points referring to constant water velocities lie in positions too erratic to enable

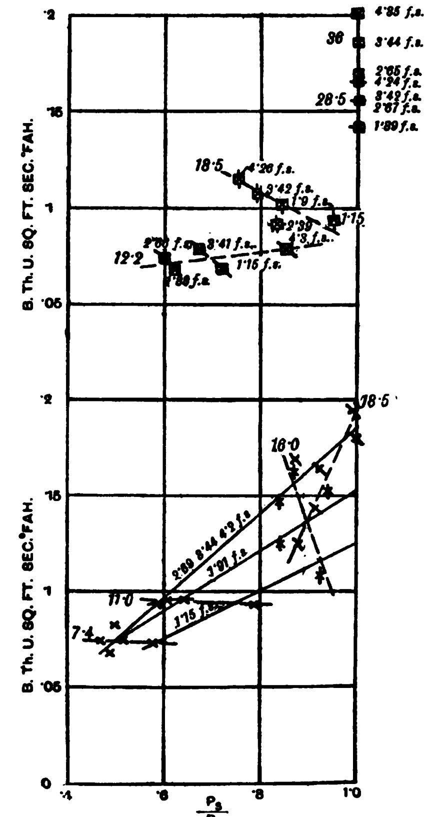


Fig. 84.—Variation of rate of heat transmission from Weighton's condenser experiments.

the general relation at constant velocity to be shown by lines, but in both cases it is seen that the lines at constant rates of condensation are very steep at high rates and are practically horizontal at low rates. This again signifies, therefore, that an increase of the velocity of flow of the water has very little influence on the rate of heat transmission at low rates of

condensation $\left(\text{low values of } \frac{P_s}{P_t}\right)$, but at high rates of condensation $\left(\text{high values of } \frac{P_s}{P_t}\right)$ the velocity of the water is an important factor.

In view of the difficulties associated with the conditions on the steam side of the condenser tubes, it is therefore not surprising to find very large differences in the rates of heat transmission from the steam to the water for the various

FIG. 85.—Rates of heat transmission in surface condensers.

condensing arrangements which have been described. This is well shown in Fig. 85, where the rates of heat transmission are compared on a basis of water velocity. Most of the curves in this figure merely indicate approximately the range between the extreme values obtained in the individual condensers.

Various writers and experimenters have attempted to express the rate of heat transmission, h, in surface condensers in terms of the water velocity v. Some have used $h \propto \sqrt{v}$ and others $h \propto \sqrt[3]{v}$. A reference to the various curves in Fig. 85, however, shows how futile are such formulæ, except perhaps for the particular apparatus and for the particular conditions of operation from which they were obtained.

With further reference to the rate of heat transmission in surface condensers Professor G. G. Stoney, in reply to the discussion on his paper, "The Effect of Vacuum in Steam Turbines," * stated that the probable rate in good average condensers for steam turbines at full load was ·17 to ·22 B.Th.U. per sq. ft. per sec. per deg. Fahr. difference between steam and water, depending upon the cleanliness of the tubes, going down perhaps to ·11 to ·14 at reduced loads.

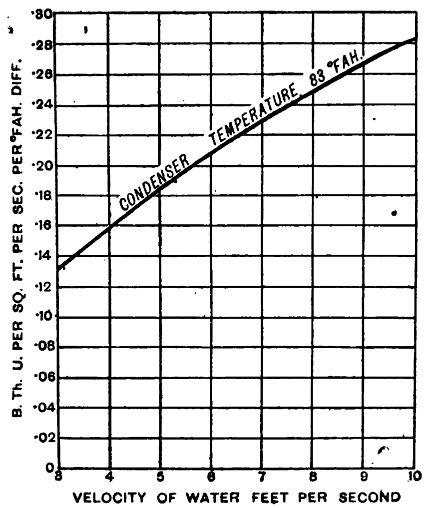


FIG. 86.—Probable rate of heat transmission in modern surface condensers at high rates of condensation and with small air leakage.

If it be taken that the velocity of the cooling water would be in the neighbourhood of 5 to 7 ft. per second, now common in condenser practice, the rates of heat transmission mentioned here are well within the limits shown in Fig. 85.

The rate of heat transmission given in Fig. 86 were stated by W. Weir† (now Lord Weir) to be applicable to Weir surface condensers under normal conditions of operation. Comparing the curve in Fig. 86 with those given in Fig. 85 it is seen

* *Journal Inst. Mech. Engs.*, Dec., 1914.

† "Development in Auxiliary Units between Exhaust Pipe and Boiler," *Trans, Inst. Engs. and Shipbds. in Scotland*, Vol. LVI, 1912–13.

that the results expressed by the curve are reasonable values with clean tubes when thorough precautions are taken to prevent excessive leakage of air into the condenser.

Feed Heater Experiments.—Many of the experiments so far discussed are applicable to surface feed heaters. In such heaters the feed water is usually pumped through the tubes on its way to the boilers and is heated by auxiliary or waste exhaust steam.

An extensive series of experiments on certain types of surface feed heaters have been made by Professor Leo. Loeb.* The first series to be discussed were made on a feed heater containing 117 semicircular $\frac{3}{4}$-in. bore tubes, No. 16 B.W.G., expanded at both ends into a horizontal brass tube plate. The tubes varied in length from 20·6 in. on the inner row of $5\frac{5}{8}$ in. radius to 69·7 in. on the outer row of 21 in. radius. The total heating surface was 88·2 sq. ft. of which 86 sq. ft. was tube area. The feed water flowed through the tubes in one pass and the steam condensed on the outside.

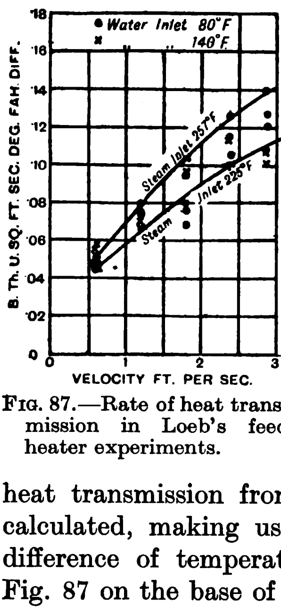


FIG. 87.—Rate of heat transmission in Loeb's feed heater experiments.

From the tabulated data in the paper the various rates of heat transmission from the steam to the water have been calculated, making use of equation 1, p. 121, for the mean difference of temperature, and these are shown plotted in Fig. 87 on the base of mean water velocity. In the first set of experiments the saturation steam inlet temperature varied in different tests from about 225° F. to about 257° F., the steam being superheated several degrees at inlet, and the water of condensation left the heater at temperatures about 10° F. below the saturation temperatures of the entering steam. To obtain consistent results steam cocks were fitted at the top and bottom of the heater and allowed to blow steam slightly to get rid of the air. The black points in Fig. 87 illustrate the results obtained with a water inlet of about 80°

* *Jour. Amer. Soc. Naval Engs.*, Vol. XXVII, 1915.

F., and the crosses with a water inlet about 140° F. The rise in the temperature of the water through the heater varied from about 20° F. to 40° F. according to circumstances. The curves indicate roughly the extreme values obtained for these tests.

Some experiments were also made with retarders inserted in the tubes consisting of annealed copper strips ⅝ in. wide and ·0268 in. thick twisted into spirals of six inches pitch. The saturated steam inlet temperature varied from about

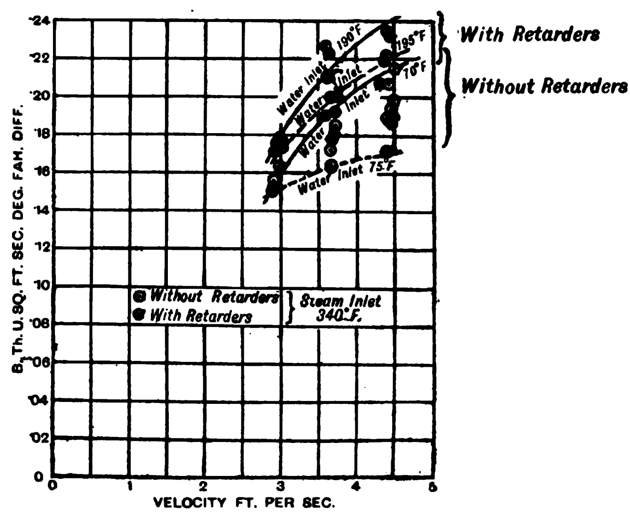


FIG. 88.—Rate of heat transmission in Loeb's feed heater experiments.

236° F. to about 260° F., the water of condensation being usually 10 to 20° F. below these temperatures. The water inlet temperatures varied from about 70° F. to 190° F. The crosses with enclosing circles in Fig. 88 represented the results and the lines drawn in are to indicate the extreme values. A similar set of tests were made without retarders over corresponding ranges of temperatures, and the results are represented by the black points with enclosing circles. A comparison between these two sets of tests showed that the retarders caused an increase in the rate of heat transmission within the ranges of the experiments. There would

also be an increase of the resistance to the flow through the tubes due to the retarders, but if placed on the delivery side of the feed pump, as is the usual practice, the small increase of the resistance is of little or no importance, especially so if the pump is steam driven, and the heat in the exhaust from the pump utilised in the heater.

Further experiments were made on a spirally corrugated film heater. A heating element in this apparatus consisted of two spirally corrugated tubes, one inside the other. The water was forced through the space between the two tubes, whilst the steam had access to the inside of the inner tube and to the outside of the outer tube, the flowing film of water being thus heated from both sides. The experimental apparatus consisted of four pairs of tubes arranged in two passes, and as originally constructed the approximate thickness of the water film was $\frac{7}{64}$ in., afterwards increased to $\frac{3}{16}$ in. by replacing the inner tubes by tubes of rather smaller diameter. The apparatus was constructed so that the inner tubes could be readily withdrawn on breaking external joints only. A large number of tests were made with both of these arrangements with the tubes nearly horizontal and also when in the vertical position. The Table 15 gives the leading dimensions :—

TABLE 15

	Outer tubes.	Inner tubes.	
		With 7/64 in. film.	With 3/16 in. film.
Length overall . . .	3 ft. 11 in.	4 ft. 10¼ in.	4 ft. 10¼ in.
Maximum inside diameter .	2·373 in.	1·997 in.	1·936 in.
Minimum ,, ,, .	1·813 in.	1·375 in.	1·193 in.
Maximum outside diameter .	2·563 in.	2·163 in.	2·102 in.
Minimum ,, ,, .	2·003 in.	1·541 in.	1·359 in.
Depth of corrugations . .	·28 in.	·311 in.	·311 in.
Pitch ,, ,, . .	1·125 in.	1·125 in.	1·125 in.
Thickness of tube, B.W.G. .	13	14	14
Total heating surface of tubes sq. ft.		21·33	20·45
Total area for water-flow between tubes (normal to axis of tubes sq. ft.		·0103	·0159
Length of heater overall		6 ft. 9¼ in.	6 ft. 9¼ in.
External diameter of shell		10¾ in.	10¾ in.

In these tests the steam at inlet was always superheated by from 20° F. to 60° F. with the $\frac{7}{64}$ in. film apparatus and by

from 20° F. to 80° F. with the $\frac{3}{16}$ in. film. The water of condensation left the heater at a temperature only three or four degrees Fahr. at most below the inlet saturation temperature, passing into a drain pot from which the water was cooled and weighed. The air was liberated from this drain pot at a pet cock. The calculated rates of heat transmission are shown plotted on the base of water velocity* in Fig. 89 and it would

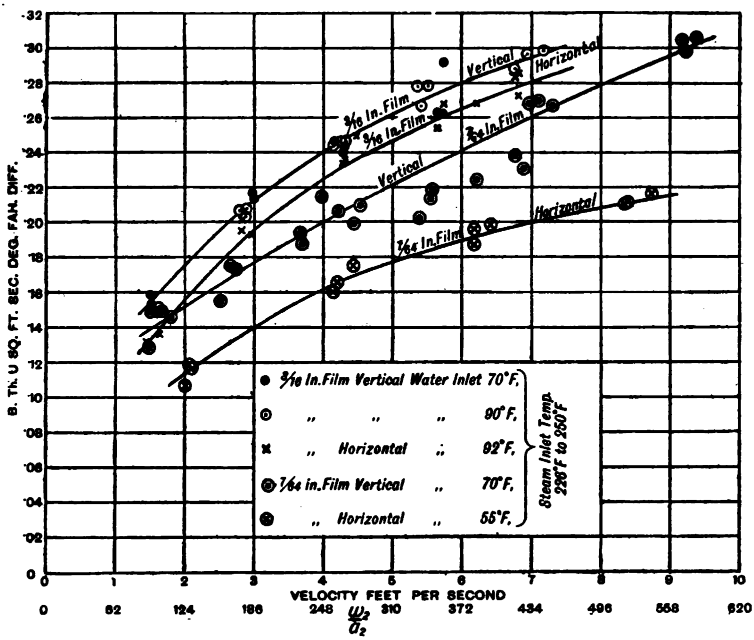


FIG. 89.—Rate of heat transmission in Loeb's experiments with feed heaters of the film type.

be seen that the vertical position gave higher rates than the horizontal in both cases, possibly due to the better manner in which the air could be liberated when in the vertical position and the easier drainage of the inside corrugated tubes.

To show the influence of air accumulation on the steam side of the tubes a run was made of about three hours' duration on the apparatus ($\frac{7}{64}$ in. film) in the vertical position. During

* This is the calculated mean velocity measured parallel to the axis of the tubes. The actual velocity was probably greater than this, as no doubt some of the water would follow the spiral corrugations.

this period no air was liberated from the apparatus except such as might be entrained in the water of condensation. The following results have been taken from the series of observations made.

Time hr. min.	Water temperatures.		Saturation steam temperature at inlet. °F.	Water of condensation. °F.	B.Th.U. per sq. ft. per sec. per deg. Fahr. temp. diff.
	Inlet.	Outlet.			
1–4	61·9	194·4	239·4	230·9	·184
4–0	62·2	181·8	239·4	216·1	·155

In these film heaters the resistance offered to the flow of the water between the inlet and outlet pipes was measured by means of a mercury U-gauge. The writer has plotted the logarithms of the pressure difference between inlet and outlet against the logarithms of the velocity of flow measured parallel to the axis of the tubes. Expressing the drop of pressure δP in lbs. per sq. in., and the velocity v in feet per second, the following results were deduced :—

$\frac{7}{64}$ in. film	Vertical	$\delta P = \cdot 912\, v^{1\cdot 5}$	(water inlet 70° F.)
	Horizontal	$\delta P = \cdot 620\, v^{1\cdot 81}$	(„ 55° F.)
$\frac{3}{16}$ in. film	Vertical	$\delta P = \cdot 617\, v^{1\cdot 76}$	(„ 70° F.)
	Vertical	$\delta P = \cdot 564\, v^{1\cdot 745}$	(„ 90° F.)
	Horizontal	$\delta P = \cdot 564\, v^{1\cdot 745}$	(„ 92° F.)

It will be noted that the law of resistance for the $\frac{3}{16}$ in. film comes out more consistent than for the $\frac{7}{64}$ in. film. An estimate of the coefficients of resistance might be made but would hardly be of much value seeing that the character of the flow and the real path of the water between the tubes is unknown. In any case, with a feed heater placed on the delivery side of the feed pump, the resistance to the flow of the water through the tubes is not, within limits, a matter of great importance.

A similar but less extensive series of experiments are recorded in the same paper made on a smaller heater of the film type with water and also with oil flowing between the tubes, The discussion of these experiments is deferred to p. 204,

Relations between Steam and Tube Temperatures and Rate of Heat Transmission.—The large variations in the rate of heat transmission from the steam to the water through a tube surface, exhibited by the results in Fig. 85, shows how desirable it is to have some means of separating the influence of the conditions on the steam side from those on the water side of a tube with greater certainty than has been possible in the previous discussion. From the results of Dr. T. E. Stanton's experiments discussed on the p. 117 as below, it is to some extent possible to calculate the approximate temperature of the tube under ordinary condenser conditions. In his paper on "The Efficiency and Design of Surface Condensers"* Dr. Stanton has applied his calculations to certain experiments made by him on surface condensers, and it is proposed here to consider the results obtained.

In the first place the formula 8 on p. 117 of *Heat Transmission by Radiation, Conduction, and Convection* was modified to the more approximate form,

$$\dagger Kl=\frac{d(930vd)^{2-n}}{1+a\left(\frac{\theta_w+t_m}{2}\right)}\log_e\frac{\theta_w-t_1}{\theta_w-t_2} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Where, ‡l=effective length of tube, ft.

d=diameter of tube on water side, ft.

v=mean velocity of water through tube, ft. per sec.

θ_w=temperature of tube, deg. Fahr. (water side), assumed to be constant.

t_1=initial temperature of the water, deg. Fahr.

t_2=final temperature of the water, deg. Fahr.

t_m=mean temperature of the water, deg. Fahr.

and K, n, and a constants which depend upon the condition

* *Proc. Inst. C.E.*, Vol. CXXXVI, 1898–99, Part II, p. 321.

† It should be noted that in this and in the following formulæ the water is taken to flow through the tube and the steam to condense on the outside.

‡ The effective length is the total length of the tubular waterway. Thus in a condenser with tubes 7 ft. long in which the water circulates in one direction through half the tubes and returns through the other half, that is, two passes, the effective length is $2\times7=14$ ft.

of the metal surfaces on the water side, the given values for smooth brass pipes being,

$$\left.\begin{array}{l} K = \cdot 0105 \\ n = 1\cdot 86 \\ a = \cdot 004 \end{array}\right\} \text{from previous experiments.}$$

Some experiments were made by Dr. Stanton on a surface condenser at the University College, Liverpool. The condenser had 152 brass tubes, $\frac{5}{8}$ in. outside diameter, 7 ft. long, and placed horizontally, the effective length being 14 ft. The

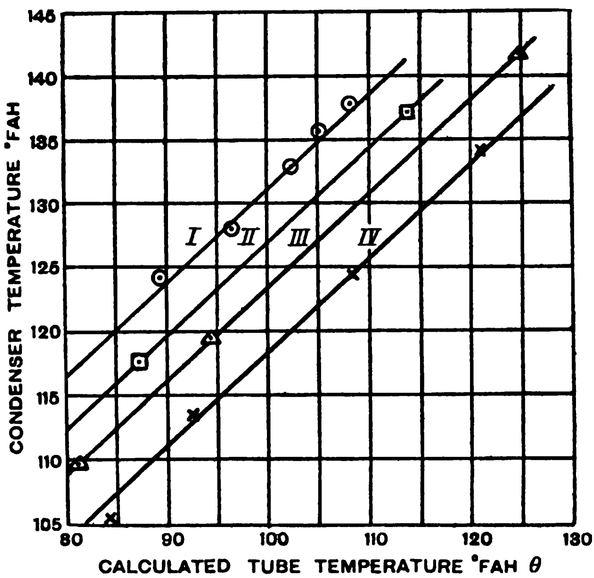


FIG. 90.—Relation between condenser temperature and calculated tube temperature, from Stanton's experiments.

flow of circulating water was estimated by a float in a tank provided with a rectangular notch.

Observations were made of the actual temperatures, and of the pressures, as shown by a mercury gauge, of the steam in the condenser at different rates of flow of the circulating water. It was found that the observed steam temperatures were in general agreement with those corresponding to the pressures. The values of θ_w were then calculated by the substitution of the experimental values in equation 1.

Some of the results are shown plotted in Fig. 90, curves III and IV. It is seen that these lines are straight and parallel, showing that the relation between the steam temperature, T_s, and the calculated surface temperature, θ_w, may be written,

$$T_s = m\theta_w + c \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

In these experiments m was said to have the value ·782,*

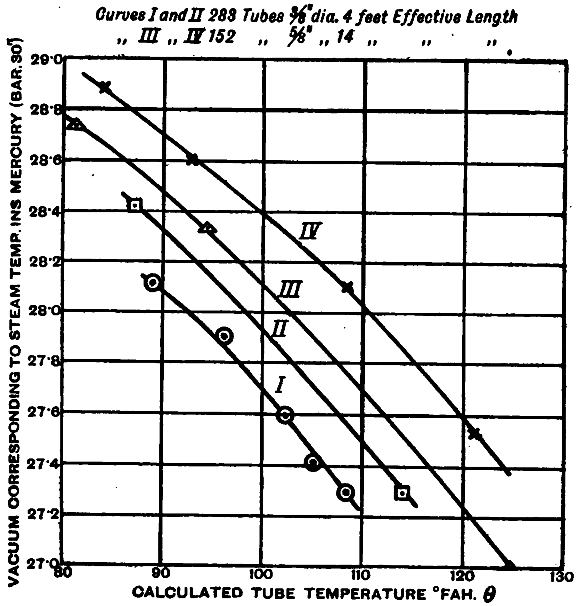


FIG. 91.—Relation between vacuum and calculated tube temperature from Stanton's experiments.

the value of the constant c depending upon the speed of the air-pump and the quantity of steam condensed. At the normal speed of the air-pump, when condensing 28 lb. of steam per minute, the value of c was 46·5 with the particular conditions obtaining in the condenser.

The curves I and II of Fig. 90 refer to experiments which were made for Dr. Stanton on a surface condenser in the Whitworth Engineering Laboratory, Manchester University. The condenser had 283 horizontal brass tubes, ⅝ in. diameter,

* On trial the writer found that the average value of m from the lines in the figure came out about ·74 instead of the value ·782.

4 ft. long, the effective length being 4 ft. as there was only one pass, and therefore the conditions of flow were very different from those of the preceding case. The chief point to notice is that in all the series shown in Fig. 90 the slope of the lines were independent of the conditions. It was also found that, even by assuming different values for the constants in equation 1, p. 183, the relation between the steam and the surface temperature was still linear, the value of the gradient m depending on the constants adopted.

Reading off steam temperatures from Fig. 90 and using steam tables to find the corresponding pressures, the corresponding vacua are shown plotted in Fig. 91 on the base of tube temperature, θ_w.

On the basis of equation 1, p. 183, Dr. Stanton deduced certain general equations relating to condenser conditions.

If N=number of tubes in one pass.

A=total surface of tubes in one pass, water side.

Q=flow of water, cub. ft. per sec.

Then,
$$N\pi dl=A.$$
$$\text{and, } Nv\frac{\pi d^2}{4}=Q.$$
$$\text{or, } v=\frac{4Ql}{Ad}.$$

Equation 1 may be written,
$$Kl=\left(\frac{3720Ql}{A}\right)^{2-n}\frac{d}{1+a\left(\frac{\theta_w+t_m}{2}\right)}\log_e\frac{\theta_w-t_1}{\theta_w-t_2} \quad . \quad . \quad . \quad . \quad (3)$$

Taking all the values in this equation to be constant except the tube diameter d and the temperatures, then,
$$\theta_w-t_1=(\theta_w-t_2)\epsilon^{\frac{b}{d}}$$
$$\text{where } b=\frac{Kl\left\{1+a\frac{(\theta_w+t_m)}{2}\right\}}{\left(\frac{3720Ql}{A}\right)^{2-n}}$$
$$\text{or, } \theta_w=\frac{t_2\epsilon^{\frac{b}{d}}-t_1}{\epsilon^{\frac{b}{d}}-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The tube efficiency may be expressed by :—

$\frac{t_2-t_1}{\theta_w-t_1}$, which becomes,

$$\frac{t_2-t_1}{\theta_w-t_1}=\frac{(t_2-t_1)\left(\epsilon^{\frac{b}{d}}-1\right)}{t_2\epsilon^{\frac{b}{d}}-t_1-t_1\epsilon^{\frac{b}{d}}+t_1}=1-\epsilon^{-\frac{b}{d}} \quad \ldots\ldots \quad (5)$$

An equation which shows that $w(t_2-t_1)$, which is the heat abstracted, has its greatest value when d is very small.

As a second example, the effect of the length of the tubes may be found for a condenser of given cooling surface, diameter of tubes and supply of water. In this case the equation for tube efficiency becomes,

$$\frac{t_2-t_1}{\theta_w-t_1}=1-\epsilon^{-b'l^{n-1}} \quad \ldots\ldots \quad (6)$$

where b' is a constant, which shows that (t_2-t_1) has its greatest value when l is greatest.

Again, if the cooling surface, length, and diameter of tubes are fixed, the tube efficiency may be written in the form,

$$\frac{t_2-t_1}{\theta_w-t_1}=1-\epsilon^{-\frac{b''}{v^{2-n}}}, \text{ where } b'' \text{ is a constant} \quad . \; . \quad (7)$$

If, ρ=density of water.

a=total area of flow.

$w=\rho av$.

$$\text{Then, Heat abstracted}=\rho av(t_2-t_1)$$
$$=\rho av\,(\theta_w-t_1)\left(1-\epsilon^{\frac{-b''}{v^{2-n}}}\right) \quad . \; . \; . \; . \quad (8)$$

Now, since the value of $(2-n)$ is about ·14, the variation in the value of $\left(1-\epsilon^{\frac{-b''}{v^{2-n}}}\right)$ will be small, so that within limits the total heat abstracted should be very nearly proportional to the velocity v, i.e. to the quantity of circulating water.

It would be noted that in equation 3, p. 186, when Q and A are constant, the condition of equally effective transmission of heat for different tube arrangements is—

$$\frac{l^{(n-1)}}{d}=\text{constant} \quad \ldots\ldots \quad (9)$$

Hence, if this ratio be determined for a given condenser of known effective performance, the length of the tubes in an equally efficient condenser of the same cooling surface made with tubes of different diameters can be at once determined.

In practice the diameter of tubes should be chosen which offers the least resistance to the flow of the water, all other things equal. Now the resistance in a tube is proportional to $\frac{lv^n}{d^{3-n}}$ for water at a given temperature; and under the specified conditions that the supply of cooling water Q and the total surface A are constants, v is equal to $\frac{Q}{\frac{\pi}{4}d^2N} = \frac{Ql}{\pi dlN\frac{1}{4}d} = \frac{4Ql}{Ad}$, that is, v is proportional to $\frac{l}{d}$. Hence the frictional resistance is proportional to

$$\frac{l}{d^{(3-n)}}\left(\frac{l}{d}\right)^n \propto \frac{l^{(n+1)}}{d^3}.$$

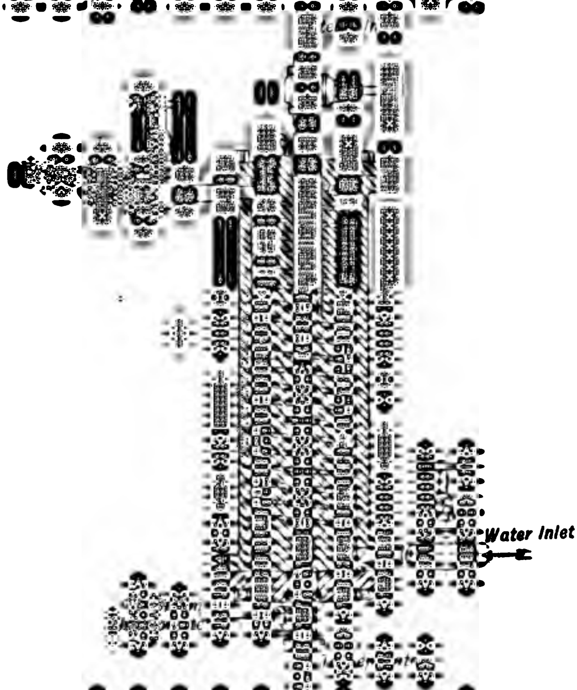
If $\frac{l^{(n-1)}}{d}$ is made constant (see equation 9).

Then, $\frac{l^{(n+1)}}{d^3} = \frac{l^{(3n-3)}}{d^3} . l^{(4-2n} \propto l^{(4-2n)}$.

Since the value of n above the critical velocity is usually somewhat less than 2, but greater than 1, except perhaps when the pipe is very rough, when $n=2$, it follows that the resistance decreases with l, except in the limiting case of $n=2$. If, then, $\frac{l^{n-1}}{d}$ is constant under the conditions specified above, the value of d decreases with l and therefore it follows that the total resistance usually decreases as d decreases when Q and A are maintained constant. The resistance to the flow of the water is therefore least when the diameter of tube is the least practicable, and the most efficient design from the point of view of heat transmission would be secured by using small tubes of as great a length as the extreme limit of resistance allowed.

The preceding demonstrations and calculations, due to Dr. Stanton, are based on tube wall temperatures calculated

[illegible] tube. In order [illegible] steam condenses [illegible] Nicolson* made [illegible] ncipally with a [illegible] condensation in [illegible] trated in Fig. 92



[illegible] -tube

[illegible] meter and 2 ft. [illegible] Thermometers [illegible] f small diameter [illegible] d with mercury. [illegible] l hole and the [illegible] high rate. This [illegible] were formed to [illegible] nents the steam

* [illegible]," *Brit. Ass. Report* [illegible]

condensing surface was wiped continually by a revolving brush constructed of thin strips of steel so that the surface was brushed five or six times per second, with the idea of preventing any accumulation of water of condensation on the surface. It was found that even such a vigorous brushing action only increased the condensation by about 5% on the average of several experiments. Some trouble was experienced by the drilled holes disturbing the flow of heat in the metal, but otherwise the results were consistent. Equation 3, p. 24, of *Heat Transmission by Radiation, Conduction, and Convection* indicates that the relation between the metal temperature and the distance from the centre could be expressed in logarithmic form. Knowing the temperatures at definite positions, the inner and outer surface temperatures could be calculated. It was found that the rate of heat flow h_s in B.Th.U. per second per square foot of surface in contact with the steam per degree Fahr. difference between the steam (T_s) and the surface (θ_s) could be expressed by

$$h_s = \cdot 74.$$

Observations at different steam temperatures did not indicate any marked differences in the rate of heat transmission from the steam to the metal.

It was also possible to deduce the conductivity of the metal. For the cast-iron cylinder it was found to be about 5·5 B.Th.U. per sq. ft. per min. per degree Fahr. per in. thick, and for a mild steel cylinder which they also used about 5·8 to the same units, the mean temperature being about 140° F.

In the same report of the British Association Professor Callendar also describes some experiments he made where the condensing tube was platinum, $\frac{1}{4}$ in. diameter, 16 in. long, the thickness being $\frac{6}{1000}$ in. The mean temperature of the tube was determined by measuring the electrical resistance of that portion of the tube on which the steam was condensing. Since the tube was so thin the greatest difference of temperature between the outer and inner surfaces did not exceed $\frac{1}{4}$° C. The platinum tube was enclosed in an outer case of brass or one of glass, the steam was admitted to the space between the tubes and the steady current of condensing water flowed through the $\frac{1}{4}$-in. tube. The pressure of the steam

was nearly atmospheric and was measured by a mercury column. The outlet and inlet water temperatures were measured, together with the quantity flowing through the tube. The results obtained for the heat transmission from the steam to the surface were rather higher than for the thick cylinder experiments.

Summary of Results.

1. With a short length of condenser and very free escape of steam, condensation of 22·2 B.Th.U. per sq. ft. per sec. for temperature difference, $(T_s - \theta_s)$, of 28·5° F. was obtained, equivalent to ·78 $(T_s - \theta_s)$. This was the smallest rate observed, the tube being in the vertical position and the steam flow downwards.

2. With the same conditions, but having the length of tube exposed to steam nearly twice as great, condensation was obtained of 22·3 B.Th.U. per sq. ft. per sec. for a temperature difference of 25·3° F., equivalent to ·88 $(T_s - \theta_s)$. The lower half of the tube was more thickly covered with water of condensation than the upper half and the steam was full of flying spray, which may have assisted in conveying heat to the metal and in maintaining the same rate of condensation on the lower half as on the upper.

3. With the same arrangement but with the steam flow upwards and reduced until escape of steam was as gentle as possible consistent with keeping the apparatus full of steam and excluding air, the rate of condensation was somewhat larger, 23·6 B.Th.U. per sq. ft. per sec. with the temperature difference only 22·0° F., equivalent to 1·07 $(T_s - \theta_s)$. The gentle upward stream of steam tended to keep the surface covered relatively thickly with drops and rivulets of water. It would appear probable that the large surface exposed by drops was so much greater (in the present instance about twice as great) than the surface of the metal, and that the drops of water themselves were in such rapid motion, that the increased surface more than compensated for any resistance the water film may have offered to the flow of heat.

4. To verify this view the outer glass tube was replaced by a much smaller tube so as to leave very little space for the

steam current. The pressure at the inlet was about 4 in. of mercury above the atmosphere and the surface of the platinum tube was violently scoured by a spiral rush of steam and spray. Under these conditions condensation was reduced to 19·2 B.Th.U. per sq. ft. per sec. for the temperature difference of 19·8° F., equivalent to ·97 $(T_s - \theta_s)$. The effect of the energetic scouring was evident from the slight rise of the

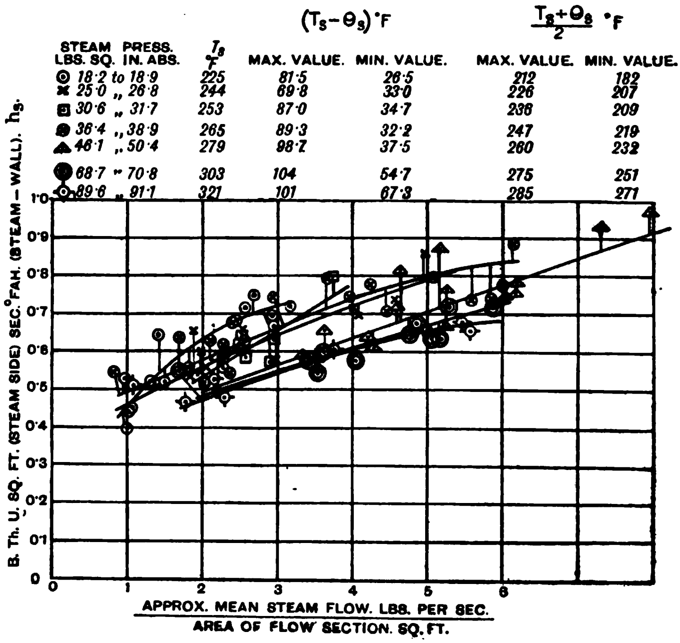


FIG. 93 —Rate of heat transmission from steam to tube, from Webster's experiments.

temperature of the metal as compared with the previous experiments.

Professor Callendar is so well known as a careful experimenter and as an expert in the measurement of temperatures by electrical methods that little doubt exists as to the accuracy of the measurements. It would therefore appear that when a tube is of small diameter and in contact with steam on the outside where there is little or no air present, the presence of moving drops of water on the tube may augment rather

than retard the rate of heat transmission from the steam to the water. This result is quite contrary to the usually accepted notions of the influence of water of condensation deposited on the tube. It would be noted in this connection that the presence of the water of condensation when condensing steam inside small tubes would probably have an opposite influence from that occurring when condensing on the outside.

In the experiments referred to on p. 141 Mr. Webster measured the tube temperatures by means of the copper-constantan thermo-junctions described on p. 122 in *The Measurement of Steady and Fluctuating Temperatures*, and the arithmetic mean temperatures on the steam side of the tube were recorded in his paper as determined from two thermo-junctions, one near each end of the tube. The tube temperatures on the water side were calculated by using the formula 3 referred to on p. 190, and were also tabulated. From the data given in the paper the writer has calculated the rates of heat transmission from the steam to the tube surface, and also from the other surface to the cooling water. The rate of heat transmission from the steam to the tube, in B.Th.U. per sq. ft. per sec. per degree Fahr. difference (steam to tube) is shown plotted in Fig. 93 on the base

$$\frac{w_s}{a_s} = \frac{\text{average weight of steam-flow past the tube, lb. per sec.}}{\text{area of steam passage, sq. ft.}}$$ the

value of w_s being taken as the arithmetic mean of the quantity of steam entering and the quantity of steam leaving the condenser. It is seen that the rate of heat transmission h_s increases appreciably with the weight of steam-flow, but decreases as the steam pressure increases. Although this latter circumstance may not have been anticipated it should be remembered that the conditions on the steam side of the tube are complicated by the possibility of the accumulation of air and water at the tube surface.

The maximum and minimum values of the temperature difference, $(T_s - \theta_s)$, and the corresponding values of $\left(\frac{T_s + \theta_s}{2}\right)$, taken to represent an average film temperature, are recorded on the diagram Fig. 93 for purposes of reference.

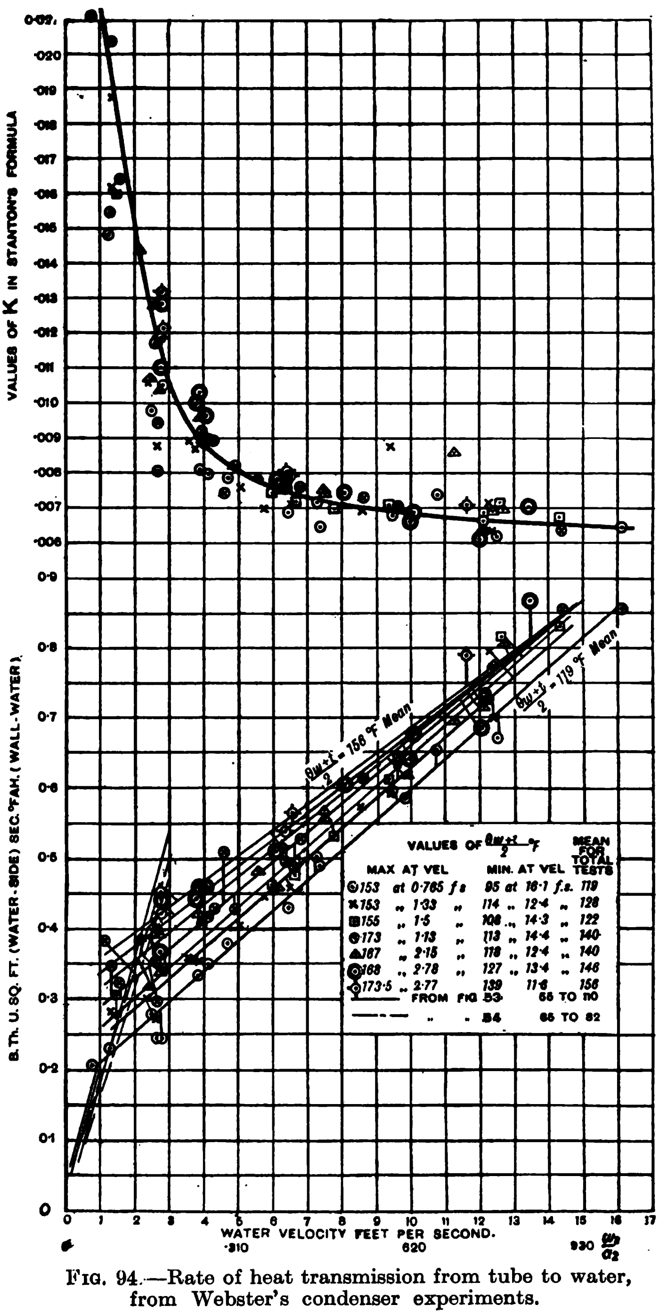


FIG. 94.—Rate of heat transmission from tube to water, from Webster's condenser experiments.

The calculated rates of heat transmission, h_w, from the tube to the water are shown plotted in Fig. 94 on the base of water velocity. Although the results are in some cases somewhat erratic it is seen that the general relation can be fairly represented by straight lines within the limits of the experiments, and that the value of h_w apparently increases appreciably at any particular water velocity as the mean temperature $\frac{\theta_w+t}{2}$ increases. Again, for purposes of reference the maximum and minimum values of $\frac{\theta_w+t}{2}$ are recorded on the diagram.

For comparison the mean straight lines from Figs. 53 and 54, pp. 138, 140 of *Heat Transmission by Radiation, Conduction, and Convection* are also reproduced in Fig. 94 on the velocity base. In these particular experiments made by Jordan the water flowed in the annular space on the outside of the copper tube with hot air flowing through the tube. The range of water velocities would be from about 0·1 to nearly 3 ft. per sec., and therefore it is probable that the critical velocity of flow was only exceeded at the higher velocities. This probably accounts to some extent for the rapid fall in the rate of heat transmission at the low velocities of flow here used.

Although the expression 1, p. 183, is only really intended to be applicable to the flow of water through a tube when the rise of water temperature is small, the writer has used it in connection with Webster's experiments, and from the known conditions of temperature, etc., on the water side of the tube has calculated the various values of K, shown plotted in Fig. 94 on the water velocity base. With such conditions it is seen that K varies considerably with the velocity of the water at low velocities but attains nearly a constant value at high velocities.

The distribution of temperature from the steam to the water is illustrated in Fig. 95 for certain of Webster's tests. The rates of heat transmission, the values of $\frac{w_s}{a_s}$, water velocity, and $\frac{w_2}{a_2}$ are given. The full lines refer to a comparatively low

rate of heat transmission and the dotted lines to a higher rate. In all these cases the arithmetic mean of the inlet and outlet temperatures of the water have been taken to represent the mean water temperatures. Although this is not quite the true mean temperature the error involved in these experiments is small and of little importance.

From the results of Messrs. Clement and Garland's experiments referred to on p. 138 the writer has also calculated the

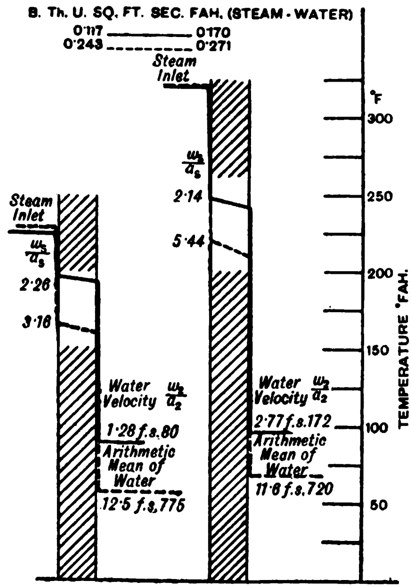


FIG. 95.—Distribution of temperature from steam to water. Webster's experiments.

various values of K derived from the expression 1, p. 183, and these values are shown plotted in Fig. 96 on a base of water velocity. It is interesting to note that, on the whole, they corroborate the results shown in Fig. 95 for Webster's experiments.

The values of the rate of heat transmission for these experiments from the tube to the water h_w are also shown plotted in Fig. 96 on a water velocity base for the three steam temperatures, 274° F., 307° F., and 330° F. Evidently the results

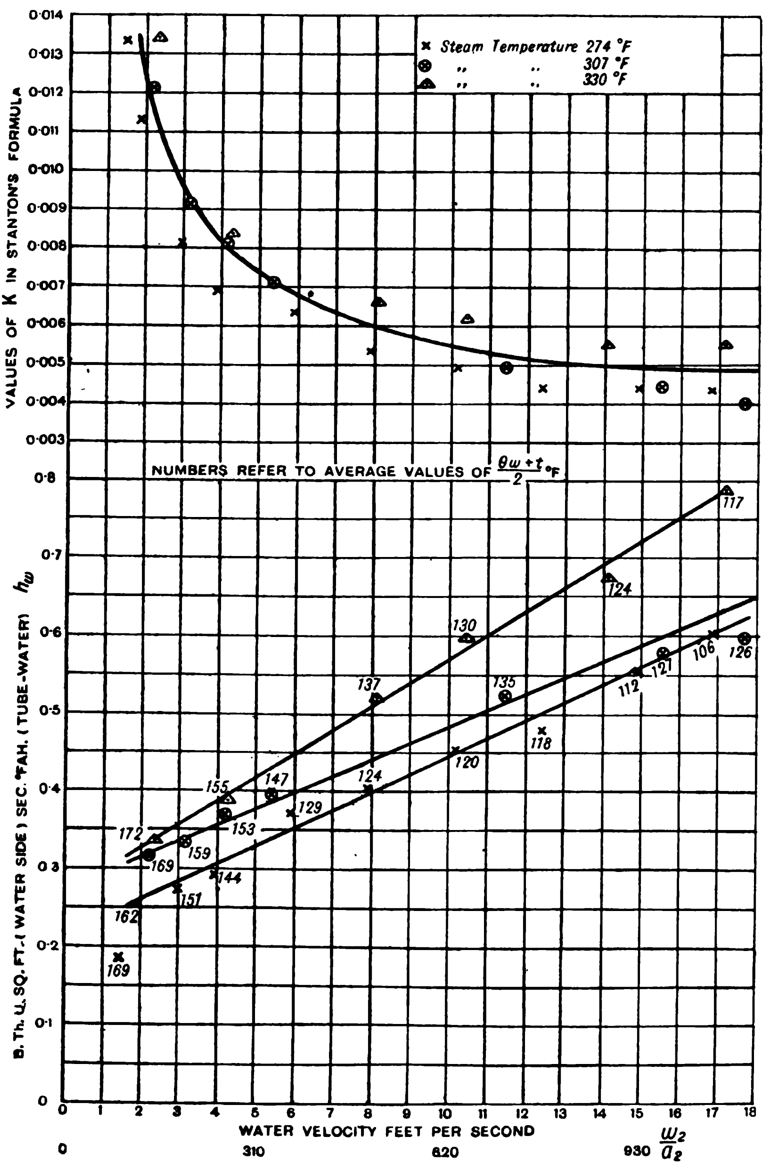


FIG 96.—Rate of heat transmission from tube to water, from Clement and Garland's condenser experiments.

for each steam temperature can be represented by a straight line. It is hardly likely, however, that the changes of the mean temperature, $\frac{\theta_w+t}{2}$, shown by the numbers on the diagram, could account for the apparent increases in the rate of heat transmission shown as the steam temperature increased.

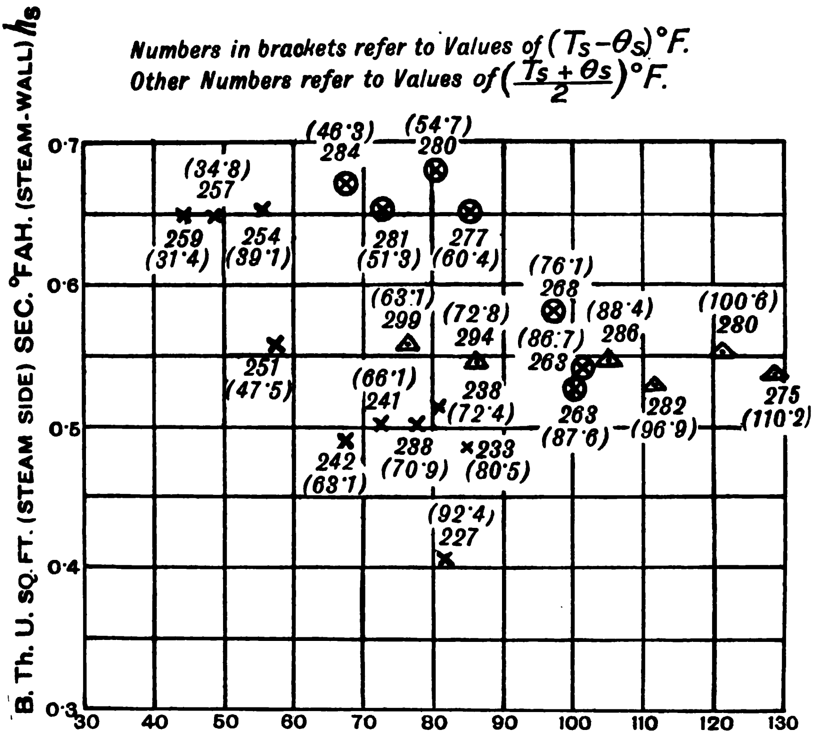


FIG 97.—Rate of heat transmission from steam to tube, from Clement and Garland's experiments.

The rates of heat transmission h_s for Clement and Garland's experiments have been plotted in Fig. 97 on the base of heat transmission. The numbers opposite the various points indicate the values of $(T_s-\theta_s)$ and $\frac{T_s+\theta_s}{2}$, but it is difficult to make comparisons with Webster's experiments because no

data is given by Clement and Garland respecting the actual weight of steam flowing through the condenser past the tube.

It is interesting to compare the various results obtained for the rate of heat transmission from steam to a tube surface. Messrs. Callendar and Nicolson, as described on p. 190, obtained a value of ·74 B.Th.U. per sq. ft. per sec. per degree Fahr. difference of temperature, when condensing inside a thick cast-iron tube of 1-in. bore. Under these conditions they found that brushing the condensing surface or altering the speed of the steam-flow had but little influence on the above rate of condensation. Condensing on a ¼-in. platinum tube, Professor Callendar obtained rates of condensation, with steam at about 212° F., varying from ·78 to 1·07 as described on p. 191, whilst he found that increasing the rate of steam-flow reduced the rate of heat transmission, for the probable reasons mentioned on p. 191. Webster's results shown in Fig. 93, p. 192, where the condensation occurred on a tube ⅞ in. outside diameter, indicate a definite increase in the rate of heat transmission on increasing the flow of steam through the condenser, and a reduction of the rate for any given weight flow of steam as the steam temperature increased. It will be noted that in most cases the values come out below ·74. Messrs. Clement and Garland's results shown in Fig. 97 are all below the Callendar and Nicolson value of ·74.

These various values, however, while useful in certain circumstances, cannot be applied to ordinary condenser conditions for estimating tube surface temperatures with any degree of certainty, because they were all obtained from experiments where the steam temperatures were much higher than in ordinary condenser practice, and they refer also to nearly air-free steam.

Although the values of K in equation 1, p. 183, derived from Webster's experiments and from Clement and Garland's experiments varied with the velocity of the water, the drop of temperature between the steam and the water and the rise of temperature of the water were all greater than obtains in ordinary condenser practice. It will therefore be taken that the constant for K on p. 184 obtained by Dr. Stanton

may be applied to ordinary condenser conditions and, within limits, may be used in the calculation of tube temperatures.

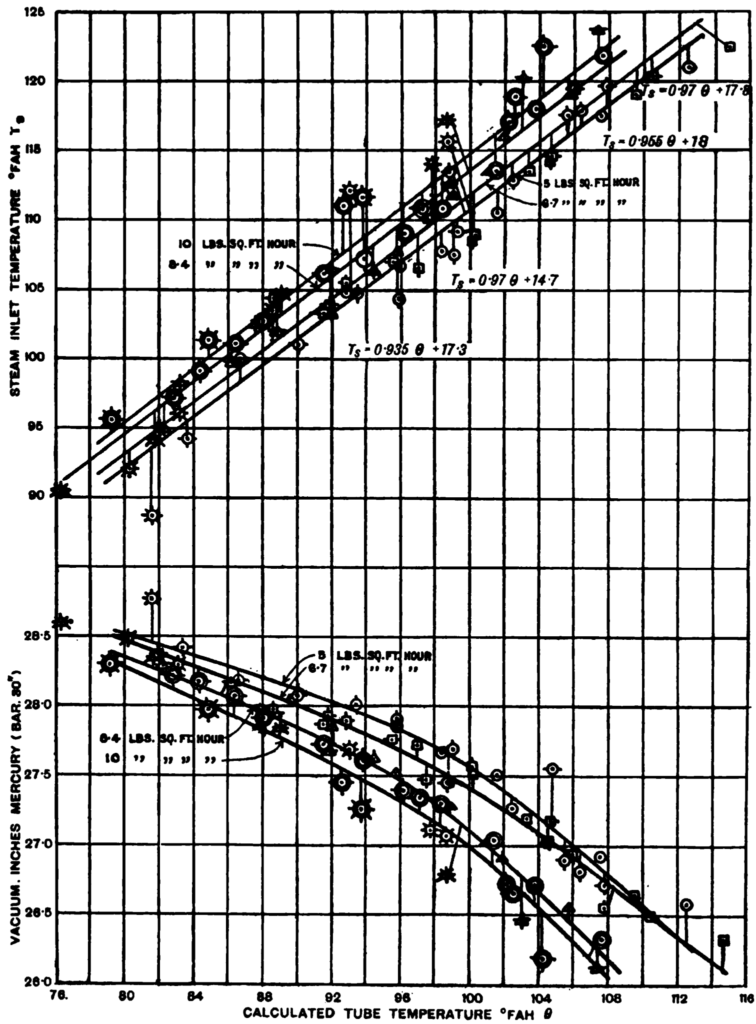


FIG. 98.—Relations between vacuum, steam temperature, and calculated tube temperature, from Allen's condenser experiments.

The writer has calculated tube temperatures in connection with Allen's condenser experiments, and although the results of the calculations show rather erratic tendencies when plotted

as in Fig. 98 it is seen that the relation between the measured steam temperature at the condenser inlet, T_s, and the calculated tube temperature, θ, may be roughly represented by straight lines as in Dr. Stanton's experiments on p. 184, Fig. 90, and that therefore,

$$T_s = m\theta + c$$

where c increases with the rate of condensation.

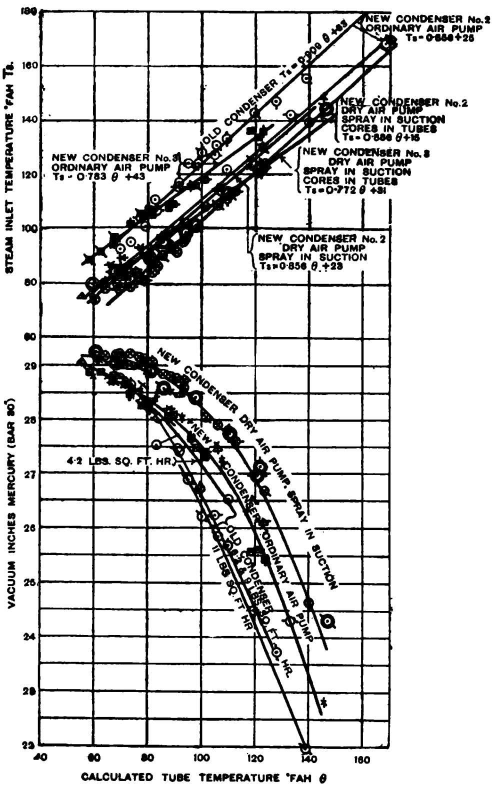


FIG. 99.—Relations between vacuum, steam temperature, and calculated tube temperature, from Weighton's condenser experiments.

The measured vacuum (barometer 30 in.) has also been plotted in Fig. 98 against the tube temperature and this indicates that the greater the rate of condensation the lower is the vacuum for any given tube temperature. Probably the

results depend to some extent upon the air-pump capacity. Mr. Allen reported that the air-pump capacity was always arranged in his experiments to have approximately a capacity of ·75 cub. ft. per lb. of steam condensed.

The writer has also calculated the tube temperatures for Professor Weighton's condenser experiments, and the measured steam inlet temperatures are shown plotted against the calculated tube temperatures in Fig. 99. Although the mean straight lines drawn can be taken to represent the relations between the steam and the tube temperatures, it may be noted that the calculated tube temperature comes out approximately the same as the steam temperature at the higher temperatures shown. This is probably due to the formula and the constants on p. 184 used to calculate tube temperatures being only really applicable to ordinary condenser conditions within a limited range. Assuming, however, that these tube temperatures are suitably calculated for the purpose in view, the relation between the tube temperature and the measured vacuum (barometer 30 in.) at the top of the condenser is also shown in Fig. 99 for the three condensers experimented upon by Professor Weighton. It will be noted that the results for the old form of condenser are of similar character to those from Stanton's and from Allen's experiments, Figs. 91 and 98 respectively. With both the No. 2 and the No. 3 condensers, however, the rate of condensation does not appear to have any appreciable influence on the vacuum for any given tube temperature. In other words, it would appear that the amount of steam condensed was practically independent of the difference of temperature between the inlet steam and the tube, at any rate within the limits of the experiments. At first sight this seems to be an extraordinary result and merits further consideration. From what has been said previously in connection with the influence of air in condensers, an increased rate of condensation usually improves the effectiveness of the lower tubes. An increased rate of condensation also means an increase in the rate of steam-flow past the tubes, and, as shown by the results in Fig. 93, p. 192, this also generally increases the rate of heat transmission with ordinary sizes of tubes. Also the water

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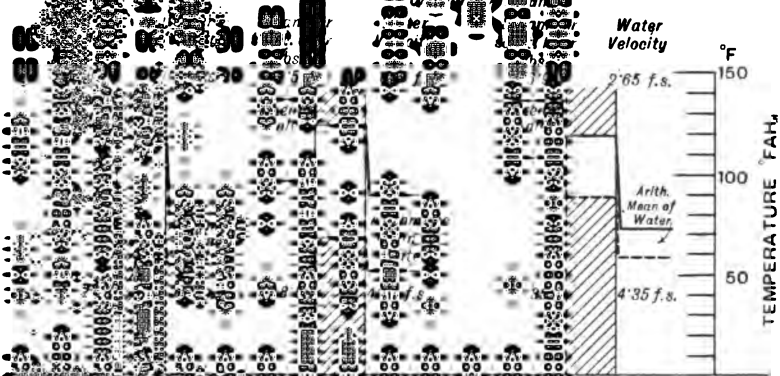

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Particulars of the rate of condensation are given in the figure.

The writer has estimated the probable tube temperatures for the film heaters described on p. 180 based upon an estimate of the difference between the temperatures of the condensing steam and the tube from Webster's experiments, p. 193, and from this has calculated the rate of heat transmission between

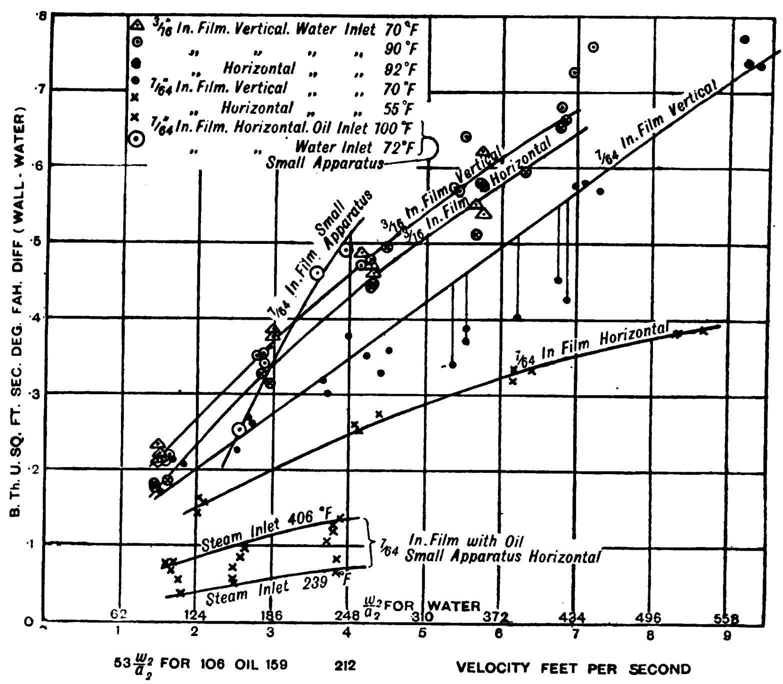

FIG. 101.—Calculated rate of heat transmission from tube to water or oil, from Loeb's heater experiments.

the tubes and the water. The values obtained are shown plotted in Fig. 101 on the base of water velocity.*

As mentioned on p. 182 some experiments were also made on a small heater of the same type having a 7/64-in. film when heating water and also when heating fuel oil. The estimated rates of heat transmission between the tube and the water and between the tube and the oil are also shown plotted in

* See remarks on p. 181.

Fig. 101. The three tests represented by the large circles refer to water heating, and the lowest series of points, indicated by crosses, showed the results when heating oil. The oil had a specific gravity ·908 at 67° F. and ·866 at 160° F. and a mean specific heat of ·476 between 212° F. and 70° F., with a flash point of 200° F. in the closed cup. It is seen that the rates of heat transmission for oil are much below the corresponding values for water at the same values of $\frac{w_2}{a_2}$. Apart from the influence of the conductivity and viscosity of the fluids, theory suggests that the rate of heat transmission should be proportional to the specific heat of the fluid, and therefore for the oil should be about half that for water. The results in Fig. 101, however, indicate that the values for the oil come out distinctly below half the corresponding values for water, and probably this is partly due to the lower conductivity and higher viscosity of the oil compared with water at the same temperature.

It is therefore obvious that where oil is heated by steam or cooled by water through a surface, the resistance to the transmission of heat is likely to be much greater between the surface and the oil than between the steam or water and the surface. Thus, to cause any appreciable increase in the rate of heat transmission of such an oil heater or cooler, it is necessary to reduce the resistance to the heat-flow between the oil and the surface either by increasing the velocity of flow or by disturbing the flow as much as possible during the transmission of heat. Thus the oil should be made to flow through small bore tubes at the highest practicable velocity, and if desirable, with retarders inserted in the tubes. The velocity of flow of the heating steam or the cooling water on the other side of the tubes is not so important as regards the transmission of heat.

Examples.—The following examples are worked out to illustrate the methods of calculation for determining the length and number of tubes required for a surface condenser under stated conditions and the resistance to the flow of the circulating water.

A steam turbine uses 30,000 lb. of steam per hour when

developing 2800 brake horse power. Steam temperature at stop valve 500° F.; saturation temperature 370° F.; exhaust steam temperature 102° F.

From Marks and Davis' steam tables :—

Heat per lb. steam at stop valve above water at exhaust temperature, 1199 B.Th.U.

$$\text{Heat supplied to turbine per second} = 1199 \times \frac{30{,}000}{3600}$$
$$= 10{,}000 \text{ B.Th.U.}$$

$$\text{Heat converted to work at shaft per second} = \frac{2800 \times 33000}{778 \times 60}$$
$$= 1980 \text{ B.Th.U.}$$

Allowing 5% heat loss by radiation, etc., from the turbine casing and the exhaust pipes at this load, then,

$$\text{Heat rejected to exhaust per second} = 10{,}000 \times \cdot 95 - 1980$$
$$= 7520 \text{ B.Th.U.,}$$
$$\text{or, } 903 \text{ B.Th.U. per lb.}$$

This, then, is the amount of latent heat carried to the condenser by the exhaust steam. Any small amount of heat given up by the cooling of the water of condensation in the condenser below exhaust temperature is neglected.

Taking the following specified conditions for the condenser:—

Brass tubes $\frac{5}{8}$ in. outside diameter, and thickness ·048 in., arranged with the same number of tubes in each of three passes. Velocity of circulating water through tubes 6 ft. per sec.

Water temperatures, inlet 75° F. and outlet 95° F. Condenser temperature, 102° F. at inlet. Vacuum, 28 in., with barometer, 30 in.

For these conditions a conservative estimate of the rate of heat transmission from the steam to the water is ·18 B.Th.U. per sq. ft. per sec. per degree Fahr. difference if reasonable precautions are taken against air leakage.

$$\text{Then, mean temperature difference} = \frac{95 - 75}{\log_e \frac{102 - 75}{102 - 95}} = 14 \cdot 8^\circ \text{ F.}$$

$$\text{Heat to water per second} = 7520 = \cdot 18 \times 14 \cdot 8 \times A.$$

$$\text{or, } A = \frac{7520}{\cdot 18 \times 14 \cdot 8} = 2820 \text{ sq. ft.}$$

$$\text{Water per second} = \frac{7520}{95-75} = 376 \text{ lb.}$$

$$\text{or, } \frac{376}{62{\cdot}2} = 6{\cdot}05 \text{ cu. ft.}$$

$$\text{Area of section of one tube} = \frac{{\cdot}785 \times ({\cdot}625 - 2 \times {\cdot}048)^2}{144}$$

$$= {\cdot}00152 \text{ sq. ft.}$$

$$\text{Then, } 6{\cdot}05 = n \times {\cdot}00152 \times 6$$

$$\text{or, } n = \frac{6{\cdot}05}{{\cdot}00152 \times 6} = 663 \text{ tubes per pass.}$$

If l'=length of each tube, ft.

$$\text{Then, } 3 \times 663 \times \pi \times \frac{{\cdot}625}{12} \times l' = 2820.$$

$$\text{or, } l' = \frac{2820 \times 12}{3 \times 663 \times \pi \times {\cdot}625}$$

$$= 8{\cdot}7 \text{ ft. between tube plates.}$$

$$\text{Steam condensed per sq. ft. per hour} = \frac{30{,}000}{2820}$$

$$= 10{\cdot}6 \text{ lb.}$$

It is sometimes desirable to check the results of such calculations by estimating the mean tube temperature, using equation 1, p. 183, and then to ascertain the probable vacuum obtainable by reference to Figs. 91, 98 and 99.

$$\text{The equation is, } Kl = \frac{d(930vd)^{2-n}}{1 + a\left(\frac{\theta + t_m}{2}\right)} \log_e \frac{\theta - t_1}{\theta - t_2}$$

Using the constants given on p. 184, that is,

$K = {\cdot}0105$, $n = 1{\cdot}86$, $a = {\cdot}004$,

l=effective length, ft., d=diam. of tube, ft.

v=water velocity, ft. per sec.

t_1 and t_2 are inlet and outlet water temperatures respectively.

$$\text{Then, } {\cdot}0105 \times 3 \times 8{\cdot}7 = \frac{{\cdot}044(930 \times 6 \times {\cdot}044)^{{\cdot}14}}{1 + {\cdot}004\left(\frac{95+85}{2}\right)} \log_e \frac{\theta - 75}{\theta - 95}$$

where θ in $a\left(\frac{\theta + t_m}{2}\right)$ has been assumed to be about 95° F.,

since any small error at this point has very little influence on the calculation.

$$\therefore \log_e \frac{\theta-75}{\theta-95} = \frac{\cdot 0105 \times 3 \times 8{\cdot}7 \times 1{\cdot}36.}{\cdot 044 \times 2{\cdot}16.}$$
$$= 3{\cdot}92.$$
$$\text{or, } \frac{\theta-75}{\theta-95} = 50{\cdot}4.$$
$$\theta = 95{\cdot}4^\circ \text{ F.}$$

If, instead of using the above value of K, the mean value from Figs. 94 and 96, ·007, say, is used in the above equation,

Then, $\theta = 96{\cdot}5^\circ$ F.

It is seen, therefore, that the value of K does not greatly affect the calculated mean tube temperature.

Taking the mean tube temperature to be about 96° F. Table 16 shows the vacuum obtainable from some of the condensers discussed previously.

TABLE 16

From Figure.	Calculated tube temperature. °F.	Vacuum, in mercury. (Bar. 30 in.).
Fig. 91, p. 185 Mean of Curves III and IV	96° F.	28·4
Fig. 98, p. 200	96° F.	27·3
Fig. 99, p. 201 Old condenser	96° F.	27·0
Fig. 99, p. 201 New condenser, ordinary air-pump . . .	96° F.	27·75
Fig. 99, p. 201 New condenser, with dry air-pump and cold water spray	96° F.	28·5

With old types of condensers, to which the first three lines in Table 16 refer, the vacuum depends upon the rate of condensation and upon the capacity of the air-pumps. In the condensers experimented with by Professor Weighton the vacuum was nearly independent of the rate of condensation.

It is therefore seen that the vacuum of 28 inches for the condenser example on p. 206 is well within the possibilities of an efficiently designed condenser where reasonable precautions are taken against air leakage.

For the sake of illustration the writer has also made calculations similar to those preceding with a view to showing the influence of the circulating water temperatures on the cooling surface required and on the length of the tubes. The results are all tabulated in Table 17. Calculations were also made of the mean tube temperature for the conditions in line 3 of Table 17. On referring to Fig. 99, p. 201, it will be seen that at about 91° F. tube temperature the vacua obtained in Professor Weighton's tests for the new condensers were 28 in. with ordinary air-pumps, and 28·6 in. with dry air-pumps and cold water spray in the air-pump suction.

Exhaust steam is sometimes used to heat the boiler feed water. The calculations relating to the design of feed heaters of the surface type are similar in character to those for surface condensers, and therefore it is hardly necessary to work out examples in this case.

Circulating pumps for surface condensers are usually of the centrifugal type. The action and design of such pumps is treated in several* text-books and papers to which reference should be made for detailed information. Briefly stated the action is as follows : The water is supplied on the suction side to the centre of the pump casing and in passing through the rotating impeller the velocity of the water and its pressure increase. Before the water is discharged a portion of the velocity energy has been converted into pressure energy in the diffusing chamber in the casing round the perimeter of the impeller. The total increase of pressure has to be sufficient to overcome the external pipe resistance and any lift or head of water that may be encountered. Such a pump will lift water by suction but it is necessary to prime first with water so as to fill the suction pipe and pump casing, and to facilitate this there should be a non-return valve at the bottom of the

* For example: *Hydraulics and its Applications*, by Professor A. H. Gibson; *Centrifugal Pumps, Turbines, and Water Motors*, by C. H. Innes. "Notes on the Construction of Turbine Pumps," by A. E. L. Chorlton, *Journal Inst. Mech. Engs.*, May, 1917,

TABLE 17

Steam condensed lbs. per hour.	Steam condensed per sq. ft. per hour. lbs.	Vacuum (bar. 30 in.).	Condenser inlet temp. °F.	Circulating water. Temperatures. Inlet. °F.	Circulating water. Temperatures. Outlet. °F.	Circulating water. lbs. per sec.	Circulating water. / Steam condensed.	Water velocity ft. per sec.	No. of passes.	Length of each tube, feet.	No. of tubes per pass.	Total cooling surface sq. ft.	B.Th.U. per sq. ft. per sec. per °F. diff.	Calculated tube temp. °F.
30,000	10·6	28 in.	102	75	95	376	45	6	3	8·7	663	2820	·18	95·4 with K = ·0105 96·5 ,, K = ·007
30,000	13·3	28 in.	102	75	90	501	60	6	3	5·2	882	2260	·18	—
30,000	18·5	28 in.	102	50	90	188	22·6	6	3	10·0	330	1620	·17	90·5 with K = ·0105 92·2 ,, K = ·007
30,000	21·3	28 in.	102	50	85	215	25·8	6	3	7·6	378	1410	·17	—

suction pipe. When water has to be lifted to a considerable height centrifugal pumps may be used in series, and this combination is then termed a multiple-stage or multiple-chamber pump, or is sometimes referred to as a turbine pump.

The end of the circulating water suction pipe in the tank or reservoir should be placed well below the surface so as to withdraw the colder water and also to prevent air from being drawn in. The end is usually fitted with a short vertical length of pipe, somewhat larger in diameter than the pipe itself, and perforated to prevent large solid bodies entering and being carried to the condenser. This perforated pipe is sometimes called a " snore " pipe. Trouble sometimes occurs by some of the perforations getting blocked by sediment, dirt, leaves, weeds, etc., and may then be somewhat difficult to clear. In a large power station taking cooling water from a river or canal the water intake is made large, and in some cases a self-cleaning rotary screen is provided to keep out weeds, leaves, etc.

The following calculations illustrate the method of estimating approximately the total resistance offered to the flow of the circulating water in a condenser plant.

Condenser.—Total length of tubes per pass 7 ft.; number of passes, 3; diameter of tubes, $\frac{5}{8}$ in. outside; thickness of tube ·048 in.; velocity of water through tubes, 6 ft. per sec. Diameter of circulating pipe, 9 in.; velocity in pipe, 7 ft. per sec.; number of right-angle bends in pipe, 6.

Case 1—Delivery and suction pipes well below the water-level in the same reservoir, therefore no work is spent in raising the circulating water. Length of straight pipe, say, 50 ft.

Case 2—The circulating water is raised to a height of 25 feet above the water level on the suction side, say, to the top of a cooling tower. Total length of pipe, say, 100 feet.

Case 1—Taking the coefficient of resistance f for the condenser tubes at ·013, and using equation 18, p. 82, of *Heat Transmission by Radiation, Conduction, and Convection.*

(a) $$y=\frac{flv^2}{2gm}=\frac{\cdot013\times6^2\times21}{64\cdot4\times\frac{\cdot53}{4\times12}}=13\cdot8 \text{ feet of water.}$$

(b) Pressure difference to produce kinetic energy at tube inlet (assuming the initial velocity negligible),

$$\frac{v^2}{2g}=\frac{6^2}{64\cdot4}=\cdot56 \text{ feet of water at each pass.}$$

$$\text{or, } =3\times\cdot56=1\cdot7 \text{ feet of water for 3 passes.}$$

It is assumed that the kinetic energy at the tube outlet at each pass is wasted.

(c) Loss of head at each tube inlet due to vena-contracta (unless entrance is well rounded),

$$=\frac{v^2}{2g}=\frac{6^2}{64\cdot4}=\cdot56 \text{ feet at each pass.}$$

$$\text{or, } 3\times\cdot56=1\cdot7 \text{ feet for 3 passes.}$$

Then, Total loss of head at condenser,

$$=13\cdot8+1\cdot7+1\cdot7=17\cdot2 \text{ feet of water.}$$

(d) Taking the coefficient of resistance f for a 9-in. pipe after lengthy service to be $\cdot011$,

$$\text{Then, } y=\frac{\cdot011\times50\times7^2}{64\cdot4\times\frac{9}{4\times12}}=2\cdot23 \text{ feet of water.}$$

(e) Assuming the loss of head at a smooth right-angle bend is equal to 10 diameters of length of straight pipe,

$$\text{Then, } y_B=\frac{6\times10\times\frac{9}{12}}{50}\times2\cdot23=2\cdot0 \text{ feet of water.}$$

(f) Pressure to produce kinetic energy at pipe inlet,

$$\frac{v^2}{2g}=\frac{7^2}{64\cdot4}=\cdot76 \text{ feet of water.}$$

(g) Loss of head at pipe inlet (assumed)

$$\frac{\cdot5v^2}{2g}=\frac{\cdot5\times7^2}{64\cdot4}=\cdot38 \text{ feet of water.}$$

(h) It is taken that none of the kinetic energy of flow at the pipe outlet is recovered.

Then, Total resistance $=17\cdot2+2\cdot23+2\cdot0+\cdot76+\cdot38$

$=22\cdot6$ feet of water.

Case 2—The condenser resistance will be the same as before, viz. 17·2 feet of water.

Resistance of 100 feet of straight pipes $=4\cdot5$ feet.

Resistance of 6 bends, 2·0 feet.

Pressure drop at pipe inlet $\cdot76+\cdot38=1\cdot14$ feet.

Then, Total pipe resistance $=4{\cdot}5+2{\cdot}0+1{\cdot}14$
$=7{\cdot}64$ feet.

Increase of head at cooling tower, 25 feet.

Total resistance $=17{\cdot}2+7{\cdot}64+25$
$=49{\cdot}8$ feet of water.

The total resistance in feet of water represents the number of foot-pounds of work which the pump has to do on each pound of water pumped. If the pump efficiency is ·6 and the direct coupled motor efficiency ·85, say, and W pounds of water is delivered per minute against a total resistance y feet of water,

$$\text{Then, Electrical horse power} = \frac{Wy}{33000 \times {\cdot}6 \times {\cdot}85}$$
$$= \frac{1{\cdot}96}{33000} Wy.$$

In considering the complete design of a surface condenser and the amount of water required, some regard has to be paid to the conditions of operation. As shown by the preceding examples, when a cooling tower is used the work done by the circulating pumps has to be much greater than when the circulating water is drawn from and returned directly to a reservoir on the syphon system. Therefore in the former case it might be economical to have a smaller velocity of flow through the condenser and a correspondingly larger condensing surface. The most economical arrangement in each case, however, could only be determined when the whole of the important conditions of operation are known, including the capital costs, depreciation and working costs of condensers, circulating pumps, cooling towers, etc., together with the amount of circulating water available and its probable inlet temperature at the condenser.

Although condensers and air-pumps have undergone considerable developments in recent years it is not the present purpose to describe or discuss detail arrangements. Reference may be made to the author's book on *Condensers, Air-pumps, and Evaporators* for this information.

Water Cooling—If often happens in land practice that a continuously fresh supply of condensing or circulating water

is not available, and in such cases it becomes necessary to cool the water after leaving the condenser for re-use. One or other of the following arrangements are usually adopted for this purpose, viz.: (1) Cooling ponds or reservoirs, (2) Cooling towers, and (3) Sprayers. In all these cases the major portion of the cooling is due to the evaporation of some of the water. The vapour is absorbed by the surrounding air and the latent heat required for the evaporation is obtained from the water remaining, which cools in consequence.

Cooling Ponds—When a cooling pond is used the hot water is usually conducted by a shallow open channel a few feet wide to the furthest point in the pond, and the cold water is taken at a point nearest the condenser. The volume of water in the pond also forms a reserve to draw upon should the make-up supply be temporarily stopped, or in case of drought when no make-up supply is available.

Cooling Towers.—When there is not sufficient reservoir capacity or exposed area either cooling towers or sprayers are in common use. If the cooling reservoir or pond is nearly sufficient for the load a common practice is to raise the water several feet and to distribute it over a simple cooler constructed of thin boards over which the water is allowed to trickle in cascades, and placed in an exposed position so as to be acted upon by the wind. Where space is valuable, however, and cheap cooling water not otherwise available, enclosed cooling towers are commonly used, where the water is raised to a height of 25 feet or so and is then allowed to trickle or splash over the surfaces of a series of rods, boards, drain tiles or galvanised wire mats, as it descends to the tank below the tower. With chimney coolers the cooling portion of the tower is enclosed except for the openings for the air at the bottom. Above this cooling portion the tower is formed into a chimney to induce a draught or current of air through the tower, giving a total height of 60 to 70 feet above the tank. Fan coolers of the same type are more certain and more effective in action than chimney coolers. With these the air is usually forced into the tower by motor-driven fans placed at the base, and although this type of cooler requires less ground space than the chimney cooler the cost of power for the fans and the

greater cost of upkeep militates against the general adoption of the fan cooler.

Spray Cooling.—In this system spraying nozzles are fixed upon a system of pipes over the cooling pond. They are commonly placed at an elevation of several feet and the spray is allowed to fall into the pond, the object being to expose the maximum surface of water to the atmosphere so as to facilitate rapid evaporation and cooling. The water has to be supplied to the spraying pipes at a pressure of 18 feet or so of water to get effective spraying.

Reference may be made to the book on *Condensers, Air-pumps, and Evaporators* for a more complete description of these various cooling arrangements.

As mentioned previously the cooling action of the air is primarily due to the capacity of the air for the absorption of vapour, the latent heat necessary for the evaporation being obtained from the water remaining. The principles involved are illustrated by the following example :—

Air is saturated with water vapour at 90° F., the total pressure being 14·7 lb. sq. in. ; to find the weight of vapour in 1 lb. air and the total heat at 90° F. reckoned from 32° F.

From steam tables the saturated vapour pressure at 90° F. is ·696 lb. per sq. in., and thus the pressure of the air is $(14{\cdot}7 - {\cdot}696) = 14$ lb. per sq. in. nearly.

For 1 lb. of air, and taking $\frac{PV}{\tau} = 53{\cdot}18$, lb., ft., °F., units,

then, $\frac{14 \times 144 \times V}{(90+460)} = 53{\cdot}18.$

or, $V = 14{\cdot}5$ cub. ft. per lb.

At 90° F. the volume of 1 lb. saturated vapour is 469·3 cub. ft., and thus the weight of vapour per 1 lb. air

$$= \frac{14{\cdot}5}{469{\cdot}3} = {\cdot}031 \text{ lb.}$$

Calculating heat quantities from 32° F.,

Heat in 1 lb. air at 90° F. . . $= (90-32){\cdot}24$
$= 13{\cdot}9$ B.Th.U.

Sensible heat in ·031 lb. vapour . $= {\cdot}031 \times (90-32)$
$= 1{\cdot}8$ B.Th.U.

Latent heat in ·031 lb. vapour . $=1041 \times \cdot 031$
$=32{\cdot}3$ B.Th.U.

Total heat in 1 lb. air and accompanying vapour . . . . $=48$ B.Th.U.

Calculating in this manner the diagrams in Fig. 102 have been constructed showing the weight of vapour in 1 lb. of saturated

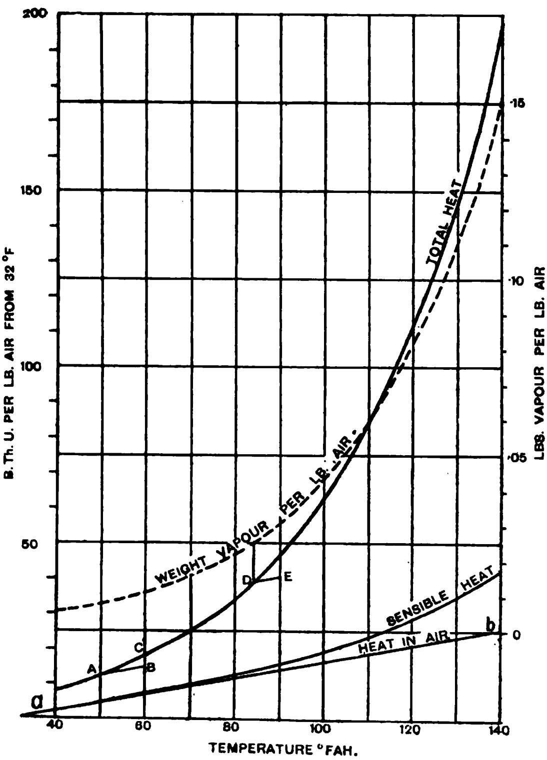

FIG. 102.—Weight of vapour and heat contents of 1 lb. air saturated with water vapour at atmospheric pressure.

air at various temperatures and the heat contents from 32° F. It will be noted how rapidly the vapour contents and the total heat increase with the temperature at the higher temperatures. An example will show how these diagrams may be utilised to estimate the cooling effect and the weight of vapour absorbed. Assuming that saturated air at 60° F. enters the base of a

cooling tower and leaves saturated at 90° F., the weight of vapour absorbed between these two temperatures is

$$\cdot 031 - \cdot 011 = \cdot 02 \text{ lb. per lb. air.}$$

The corresponding absorption of heat is

$$48 - 18 = 30 \text{ B.Th.U. per lb. air.}$$

If the air at 60° F. at the base of the tower were not saturated but had a dew-point 50° F., say, the weight of vapour at inlet is that in saturated air at 50° F. The heat contents at inlet is obtained by drawing the line A B from 50° F. parallel to the straight line *a b*, giving the point B at 60° F. This is not quite true, however, because the vapour at 60° F. is in this case superheated 10° F., but it represents too small an amount of heat to be worth considering. If the relative humidity at outlet is 80% at the outlet air temperature 90° F. then the vapour contents at outlet is

$$\cdot 031 \times \cdot 8 = \cdot 0248 \text{ lb. per lb. air,}$$

which means a dew-point temperature of about 84° F. Drawing D E parallel to *a b* gives the heat contents at the point E, again neglecting the small superheat.

Thus the absorption of vapour is

$$\cdot 0248 - \cdot 008 = \cdot 0168 \text{ lb. per lb. air,}$$

and the corresponding absorption of heat is

$$40 - 15 = 25 \text{ B.Th.U. per lb. air.}$$

If the water is cooled through, say, 30° F., then the weight of water is

$$\frac{25}{30} = \cdot 833 \text{ lb. per lb. air nearly.}$$

Thus the percentage evaporation is

$$\frac{\cdot 0168}{(\cdot 0168 + \cdot 833)} \times 100 = 2 \cdot 0\% \text{ nearly.}$$

If, say, 32 lb. of circulating or condensing water is supplied to the condenser per pound of steam condensed then the weight of air required for cooling in the tower becomes

$$\frac{32}{\cdot 833} = 38 \cdot 4 \text{ lb. per pound of water of condensation.}$$

The average percentage humidity of the air at Greenwich, Halle, and Madrid is given by Box* as follows :—

* *A Practical Treatise cn Heat.*

TABLE 18. PERCENTAGE HUMIDITY OF AIR.

Month.	Greenwich.				Halle.	Madrid.
	Temperature °F.	Humidity mean of 24 hours.	Mean of the day.		Mean of 24 hours.	
			Between 9 a.m. and 6 p.m.	Between 2 and 3 p.m.		
January	36·5	89	86	84	86	84
February	38·4	85	85	81	81	72
March	41·0	82	79	75	77	66
April	46·0	79	68	64	71	57
May	52·6	76	71	66	69	—
June	58·8	74	62	58	71	56
July	61·7	76	67	63	68	44
August	61·4	77	70	66	66	45
September	56·7	81	76	68	73	56
October	50·0	87	81	76	79	80
November	43·3	89	86	85	86	78
December	39·3	89	86	84	87	85
Mean for year		82	76	72	76	66

To obtain the humidity of the air hygrometers are used. There are various forms of these instruments and for a full description reference should be made to any standard work on the theory of heat. The hygrometer in common use, however, is known as the wet and dry bulb hygrometer, and consists of two thermometers mounted side by side, the bulb of one of these being kept wet by a wetted muslin cloth. The evaporation from the cloth causes the wet thermometer to read lower than the dry one, and on consulting Glaisher's factors (see Table 38, p. 292) the humidity is readily obtained.

In wooden cooling towers the main structure is commonly of pitch pine treated with some preservative compound such as creosote or sideroleum. All bolts and cover-plates should be galvanised, and as far as possible, nails should be avoided in fixing or supporting the internal boards or splash-bars for the reason that rusty nails cause local rotting of the wood. The enclosing boarding should be built so as to be practically free from apertures, and placed so as to give a smooth surface inside to offer a minimum resistance to the upward current of air. Fan coolers are commonly of steel, built cylindrical, with the fans at the base. As mentioned previously, the principal

cooling action is dependent upon the vapour absorbed by the air current, which really involves two more or less separate and distinct actions, viz. the heating of the air by contact with the surface of the water and the absorption of vapour by the heated air. It is obviously important to expose the maximum possible extent of water surface to the current of air. For this purpose in chimney coolers some makers distribute the water over narrow bars of wood, termed splash-bars, so that the water drops on the upper surface, whereby some of the water is split up into fine drops on rebounding from the surface, the remaining water trickling down to the underside of the bars and then falling in a similar manner on bars below, until, at the bottom of the tower, the water falls as a fine rain. Other makers depend on this action to some extent, coupled with arrangements of surfaces whereby the water is distributed in thin films as it descends over the surfaces. In any case care should be taken in the design so as not to impede uselessly the upward current of air.

The current of air through a chimney cooler is induced by the difference of density between the gases and vapour inside and the air outside the cooler. It can be shown that the theoretical velocity of flow, neglecting all frictional resistance, is given by

$$v=\sqrt{2gZ\left(\frac{\rho_0-\rho_1}{\rho}\right)} \quad \ldots \ldots \ldots \ldots \quad (1)$$

where v=velocity of flow.

g=acceleration due to gravity.

Z=mean height of travel of heated air in tower.

ρ_0=density of external atmosphere.

ρ_1=mean density of gases in tower.

$\rho=\frac{\rho_1+\rho_0}{2}$.

The above formula only applies when $\rho_0-\rho_1$ is small compared with ρ, as is the case in cooling towers. To allow for the frictional resistance of the internal fittings and sides of the cooling tower equation 1 may be modified to the form

$$v=\sqrt{\frac{2gZ}{1+F}\left(\frac{\rho_0-\rho_1}{\rho}\right)} \quad \ldots \ldots \ldots \ldots \quad (2)$$

where F is a friction constant.

If the temperature of the air and the accompanying vapour is known and the air is saturated the mean density in the tower can be calculated approximately by the same method as is given on p. 215. Similarly, the density of the outside air is calculable by means of the well-known gas law, $\frac{PV}{\tau}=c$, as is shown on p. 215.

In testing a cooler the average velocity of the current of air is usually obtained by means of an anemometer* placed in various positions in the cross-section of the cooler, but it can only give roughly approximate values.

The quantity of cooling water delivered to the tower is most easily obtained by introducing a suitable weir or notch* at the inlet to the cooler. It may also be deduced approximately by calculation from the observations on the temperatures at the inlet and outlet of the cooler in the following manner. First the evaporation per lb. of air is obtained from Fig. 102, using the observed data, and the corresponding weight of vapour evaporated per cubic foot of air is calculated. The average velocity of the air is measured by the use of an anemometer, and this, multiplied by the area of cross-section, gives the volume of the air-flow. The total volume multiplied by the vapour contents per cubic foot gives the total weight of evaporation. By consulting the data given in Fig. 102 the amount of heat required for the evaporation of this vapour is then determined, which is nearly equal to the heat given up by the cooling water. Knowing the fall of temperature of the water it is then a simple calculation to estimate the weight of water falling into the tank below the cooler. The evaporation may also be obtained when the condenser is of the surface type by shutting off the make-up supply to the tower tank and observing the fall of level in a given time when working at practically a constant load. The weight of water circulated may then be deduced as before.

Comparatively little data of a reliable nature is available from experiments on cooling towers. The following results for chimney coolers, however, seem fairly reliable within their

* Refer to *The Testing of Motive Power Engines* for a description and for the method of using this apparatus.

TABLE 19

No.	Size of tower ft.	Water lift ft.	Cubic contents cub. ft. V.	Circulating water. Gallons per hour.			Temperatures, deg. Fah.								
							Water.			Air.			Difference between air and water.		
				Normal load.	Load on test.	Per cent of normal load.	Top.	Bottom.	Drop.	Top.	Bottom.	Rise.	Top.	Bottom.	Mean.
1	35 × 20	24	*16,800*	50,000	19,000	*38*	124·5	84·0	40·5	82·2	61·6	20·6	42·3	22·4	31·4
2	,,	,,	,,	,,	24,800	*49·6*	118·4	72·6	45·8	71·4	46·7	24·7	47·0	25·9	35·5
3	57 × 24	24	*32,800*	100,000	88,900	*88·9*	91·5	75·0	16·5	73·5	48·5	25·0	18·0	26·5	22·0
4	,,	,,	,,	,,	114,000	*114*	90·9	71·4	19·5	74.6	38·9	35·7	16·3	32.5	23·6
5	72 × 25	30	*54,000*	190,000	164,000	*86·2*	96·3	82·3	14·0	84·6	58·2	26·4	11·7	24·1	17·2
6	118 × 25	26	*76,700*	250,000	119,000	*47·6*	96·0	75·0	21·0	82·6	56·3	26·3	13·4	18·7	15·9

No.	Average air velocity. ft. sec.	Evaporation. Per cent of circulating water.	Friction constant. F.	Figure of merit based upon		Probable area of water surface in contact with air, sq. ft. A.	$\frac{A}{V}$	Probable size of water-drops in tower, inches diameter.
				Net ground area.	$\frac{\text{Cubic contents.}}{100.}$			
1	3·83	3·03	10·0	*·097*	*·405*	*17,000*	*1·01*	*·014*
2	4·03	3·66	8·5	*·127*	*·53*	*22,200*	*1·32*	*·014*
3	2·91	1·16	19·7	*·136*	*·57*	*46,400*	*1·41*	*·025*
4	3·58	1·28	18·5	*·192*	*·80*	*65,500*	*2·00*	*·022*
5	3·37	1·04	13·7	*·207*	*·69*	*93,000*	*1·72*	*·023*
6	2·47	1·53	29·7	*·148*	*·57*	*109,000*	*1·42*	*·014*

limits, and were given by Mr. I. V. Robinson, Wh. Sc., in a paper* on "Cooling Towers," or are deduced from his experimental data. The values in italics were added by the writer.

The figures of merit given in Table 19 express the number of B.Th.Us. taken from the cooling water per second per degree Fahr. difference between the air and the water per sq. ft. of net ground area or per 100 cub. ft. of cubical contents. For example, in No. 1 test the heat given up by the water per second is $\frac{190{,}000 \times 40{\cdot}5}{3600} = 2140$ B.Th.U., and per degree Fahr. difference is $\frac{2140}{31{\cdot}4} = 68$ B.Th.U. This divided by the ground area, 35×20 sq. ft., or by the $\frac{\text{cubical contents}}{100} = 168$, gives figures of merit ·097 or ·405 respectively. Relatively high values would indicate that the cooler was comparatively efficient.

The rate of heat transmission between the water and the air is probably not far from the value ·004 B.Th.U. per sq. ft. of surface of contact, per sec. per degree Fahr. difference of temperature. Accepting this value for the purposes of approximate calculation it is possible to get a rough estimate of the surface of contact and of the average size of the drops or its equivalent. For example, again considering the test No. 1, the area of contact would be $\frac{68}{{\cdot}004} = 17{,}000$ sq. ft. Dividing this by the cubical contents V gives the values in the table marked $\frac{A}{V}$. Further, taking it that about 4 sec. would be required on the average for a particle of water to drop through the tower over the various splash bars or their equivalent, the weight of water falling through the air at any time would be $\frac{19{,}000 \times 10}{3600} \times 4 = 211$ lb., and the corresponding volume $\frac{211}{62} = 3{\cdot}4$ cub. ft. $= V_0$. If the number of falling drops in n and the diameter of each drop is d the total surface

* *Jour. West of Scot. Iron and Steel Inst.*, Vol. XIV, 1906–7.

$A = n\pi d^2$, assuming the drops are approximately spherical. Similarly, the volume $V_0 = n\frac{\pi}{6}d^3$.

$$\text{Thus, } \frac{V_0}{A} = \frac{d}{6}.$$

In this example,

$$\frac{V_0}{A} = \frac{3{\cdot}4}{17{,}000} = \frac{d}{6}$$

$$\text{or, } d = {\cdot}0012 \text{ ft.}$$
$$= {\cdot}014 \text{ in.}$$

From the calculated values given in Table 19 it would therefore appear that with a cooling tower working at about half the normal load the size of the drops are about ·014 in. diameter, and at about normal load ·024 in. diameter. Although the basis of this calculation of the diameter of the drops is only a very rough approximation it indicates without doubt that to get effective cooling the water needs to be split up into a fine rain or its equivalent thickness of film. To get more precise data would require a series of well-conducted experiments made first on small apparatus, where the working conditions would be easily controlled and where fairly accurate measurements were possible.

Evaporative Condensers.—An evaporative condenser usually consists of a set of tubes of brass, wrought iron, or cast iron, arranged horizontally or vertically so that the exhaust steam condenses on the inside of the tubes. Cooling water is allowed to flow over the outside of the tubes more or less in films which takes up the heat from the metal by contact. Evaporation of some of this water absorbs most of the heat as latent heat and to facilitate this evaporation the condenser is commonly placed in an exposed position so that the atmosphere and the wind take up the vapour, or else air is forced or drawn over the surface by means of fans.

Very few experimental results of a reliable nature from such condensers are available, or at any rate from which any very definite laws can be deduced. Usually the vacuum obtainable is poor when compared with ordinary surface condensers, partly on account of the great difficulty of keeping the various pipe joints air-tight, and partly because of the very large

surfaces necessary to obtain a high vacuum. Difficulties are also sometimes experienced on account of the deposits of scale on the tubes, and the design should allow of easy removal of the scale. Their main advantage when used for small reciprocating engines in place of the ordinary surface condenser and cooling tower lies in the reduced capital cost and the saving of ground space, as the condenser can often be accommodated on the roof. Also, because of the small quantity of circulating water the power absorbed by the pumps is correspondingly small. When the plant is shut down in cold weather precautions need to be taken to prevent its freezing up. At light loads the circulating water can be stopped altogether and the condensation then depends upon the cooling action of the atmosphere without evaporation.

The following data from tests made by Mr. M. Longridge* indicate the character of the results obtainable from an atmospheric evaporative condenser. The condenser consisted of two rows of vertical cast-iron tubes, ten in each row, connected at the top of each pair of tubes by a bend, and leading from one horizontal pipe into another. One of these horizontal pipes was the exhaust main from the engine and the other led to the air-pump. The vertical tubes were 4 in. diameter and about $\frac{5}{16}$ in. thick. A cast-iron trough was fixed across the row of tubes and immediately over each tube a hole in the trough allowed the water to fall directly on the centre of the bend. It then ran down the whole length of the tube into the cast-iron collecting tank below. The water was kept in circulation by means of a pump. During the test the outside of the tubes were coated with a thin scale.

Line 20 in Table 20 were calculated by the writer in the following manner, with reference to the first test.

$$\left(\begin{array}{c}\text{Steam inlet temperature} - \\ \text{mean water temperature}\end{array}\right) = 143{\cdot}6 - \left(\frac{117{\cdot}5 + 128{\cdot}4}{2}\right).$$
$$= 20{\cdot}7^\circ \text{ F.}$$

After allowing for the probable amount of water in the exhaust steam it has been taken that each pound of wet steam in condensing would give up about 900 B.Th.U.

* *Report of the Engine, Boiler, and Electrical Insurance Co.,* 1892.

$$\text{Line } 20 = \frac{485}{3600} \times 900 \times \frac{1}{272 \times 20 \cdot 7}$$
$$= \cdot 0215 \text{ B.Th.U.}$$

From this result a reference to Fig. 94, p. 194, for instance, would indicate that the average velocity of the film of water over the vertical pipes would be less than 1 ft. per sec. The writer estimates that the average thickness of the film of water on the pipes would be of the order of ·01 in. to ·02 in.,

TABLE 20

TESTS ON AN EVAPORATIVE CONDENSER HAVING VERTICAL CAST-IRON TUBES

	1892	
Date of Test	Sept. 12	Sept. 13
Weather	Wet.	Fine.
Barometric Pressure	29·8	29·5
Atmospheric temperature, °F.	—	60
External surface of condenser, sq. ft.	272	272
Duration of test, mins.	99	115
Steam condensed per hour, lbs.	485	417
Evaporation per hour, lbs.	364	334
Circulating water, lbs. per hour	6790	—
Vacuum in condenser, in. mercury	23·36	24·1
Initial temperature of circulating water in distribution trough, °F.	117·5	113·9
Final temperature of circulating water in tank below condenser, °F.	128·4	125
Temperature of make-up water from town's main, °F.	58	58
Temperature corresponding to vacuum, °F.	143·6	137
Temperature of water in hot-well, °F.	136·5	131
Steam condensed per hour per sq. ft. external surface, lbs.	1·78	1·53
Ratio of circulating water to steam condensed	14	—
Ratio of evaporation to steam condensed	·75	·80
Heat transmitted per sq. ft. per sec. B.Th.U.	·445	·384
20. B.Th.U. transmitted per sq. ft. per sec. per degree Fahr. difference between steam and water	·0215	·022
21. B.Th.U. transmitted per sq. ft. per sec. per degree Fahr. difference water and air	—	not less than ·0064

assuming the water to be fairly evenly distributed over the surfaces.

Line 21 for the second test was estimated in the following manner :—

$$\text{Mean temperature of water on pipes} = \frac{113 \cdot 9 + 125}{2} = 119 \cdot 5^\circ \text{ F.}$$

$$\left.\begin{array}{l}\text{Temperature difference between}\\ \text{water and air not greater than}\end{array}\right\} 119 \cdot 5 - 60 = 59 \cdot 5^\circ \text{ F.}$$

Therefore line 21 is not less than—

$$\frac{417}{3600} \times 900 \times \frac{1}{272 \times 59 \cdot 5} = \cdot 0064.$$

According to H. G. V. Oldham* the following results in Table 21 illustrate the average rates of condensation obtainable from evaporative condensers under ordinary conditions of operation giving a vacuum of about 24 in. of mercury (bar. 30 in.) at normal load, with a supply of circulating water from ten to fifteen times the weight of steam condensed for horizontal tubes, and about five to ten times the weight of steam condensed for vertical tubes.

TABLE 21

STEAM CONDENSED PER HOUR PER SQUARE FOOT CONDENSING SURFACE

Condensers with Horizontal Tubes, exposed to atmosphere without Fans.	Lbs. per hour per sq. ft.
Cast-iron Plain Tubes	1 to 1¼
,, ,, ,, ,, exposed to good, steady wind .	1½ to 2
Cast-iron Corrugated Tubes, Ledward's design . . .	1¼ to 1½
Wrought-iron Galvanised Tubes	2 to 2½
,, ,, ,, ,, or Copper Tubes, in good position on roof	2½ to 3
Condensers with Vertical Tubes.	
Cast-iron Plain Tubes	1½ to 2
Brass Tubes, without fan and exposed	3 to 3½
,, ,, with fan at slow speed	4 to 5
,, ,, with fan at higher speed and good water circulation	5 to 6

An experimental evaporative condenser of special design has been built and tested by J. H. Fitts.† It consisted of two rectangular end chambers connected by a series of horizontal rows of tubes, each row of tubes being immersed in a pan of water. Through the spaces between the surface of the water in each pan and the bottom of the pan above, air could be drawn by a fan. The steam inlet was connected to the top of one of the end chambers, and a horizontal diaphragm about midway caused the steam to enter the upper half of the tubes and then to flow back through the lower half. An outlet at the bottom led to the air-pump. There were 27 rows of tubes, 8 in some and 7 in others, with a total of 210 tubes. These

* "Evaporative Condensers," *Proc. Inst. Mech. Engs.*, 1899.

† *Trans. Amer. Soc. Mech. Engs.*, Vol. 14, 1892–3.

tubes were of brass, $\frac{3}{4}$ in. external diameter, No. 20 B.W.G., and 4 ft. 9½ in. long. The internal condensing surface of the tubes was 176·5 sq. ft. There were 27 cooling pans, one to each row of tubes, each 4 ft. 9½ in. by 1 ft. 9¾ in. by $1\frac{7}{16}$ in. deep. Water was fed to every third pan and 1¼ in. overflow pipes fed the rest. The total evaporating surface was 248 sq. ft. A wood casing connected one side of the condenser with a 30-in. fan, and the cooling action was obtained by the vapour absorbed by the currents of air drawn over the surfaces of the water.

For the experiments the steam supply was throttled down from a boiler, and the cooling water and condensation were measured. The following Table 22 gives an example of the results obtained.

TABLE 22

Barometer, in. mercury.	Vacuum, in. mercury.	Temperatures, °F. Steam temperature corresponding to vacuum.	Hot well.	Cooling water supply.	Water in pans above diaphragm.	Water in pans below diaphragm.	Air.	Dew-point.
28·0	16·5	167·5	149	60	140	115	70	62

Velocity of air, ft. per sec.	Volume of air moved in cub. ft. per sec.	Cooling water used per hour, lbs.	Steam condensed per hour. lbs.	Steam condensed per hour. lbs. per sq. ft. of inside tube surface.	Heat dissipated per sq. ft. of water surface per sec. B.Th.U. Top pass.	Heat dissipated per sq. ft. of water surface per sec. B.Th.U. Bottom pass.	Approximate rate of heat transmission from water to air B.Th.U. per sq. ft. per sec. per °F. between water and air. Top pass.	Approximate rate of heat transmission from water to air B.Th.U. per sq. ft. per sec. per °F. between water and air. Bottom pass.
38	108	1350*	900	5·1	1·2	·49	·023	·013

The rate of heat transmission given in Table 22 for the top and bottom passes was arrived at by the writer by a roughly approximate calculation in the following manner. In the first place it was taken that the air was evenly distributed and that an equal amount passed over the water in each pan. Taking 1 lb. air at 70° F. to have a volume 14·2 cub. ft. the weight of

* Some of this water was lost by overflow over the edges of some of the pans.

air flowing per sec. was $\frac{108}{14 \cdot 2} = 7 \cdot 64$ lb. In the top pass there were thirteen pans and in the bottom pass fourteen. Thus,

$$\text{Air through top pass} = 7 \cdot 64 \times \tfrac{13}{27} = 3 \cdot 68 \text{ lb.}$$

$$\text{Air through bottom pass} = 7 \cdot 64 \times \tfrac{14}{27} = 3.96 \text{ lb.}$$

Each pound of steam in condensing gave up about 1100 B.Th.U.

Thus, $\text{Heat per sec.} = \frac{900 \times 1100}{3600} = 275$ B.Th.U.

From a consideration of the amount of heat absorbed by 1 lb. of saturated air (see Fig. 102, p. 216) it was estimated that about ·7 of the total heat was absorbed in the top pass and about ·3 in the bottom pass.

Top Pass.—For the first approximation,

$$\text{Total evaporation in top pass} = \frac{\cdot 7 \times 275}{(1030 + 140 - 60)}$$

$$= \frac{192 \cdot 5}{1110} = \cdot 173 \text{ lb. per sec.}$$

$$\text{or,} \quad \frac{\cdot 173}{3 \cdot 68} = \cdot 047 \text{ lbs. per lb. of air.}$$

where approximately 1110 B.Th.U. were required for each pound of water evaporated. This calculation, however, does not take into account the heat absorbed in heating the air. For the second approximation the calculation then proceeds. as follows : From Fig. 102, p. 216, the vapour in each lb. air at 70° F. with dew point 62° F. is ·012 lbs.

Thus, Total vapour per lb. air $= \cdot 047 + \cdot 012 = \cdot 059$ lb.

Again from Fig. 102 the corresponding outlet temperature of the air is 110° F. Allowing for the heating of the air the probable temperature will be about 106° F., for,

Heat to air $= 3 \cdot 68 \times \cdot 24 \times (106 - 70) = 32$ B.Th.U.

Heat available per sec. for heating water and evaporation

$$= 192 \cdot 5 - 32$$

$$= 160 \cdot 5 \text{ B.Th.U.}$$

$$\therefore \text{ Evaporation} = \frac{160 \cdot 5}{1110} = \cdot 1445 \text{ lb. per sec.}$$

$$\text{or,} \quad = \frac{\cdot 1445}{3 \cdot 68} = \cdot 0393 \text{ lb. per lb. air.}$$

Total vapour $= \cdot 0393 + \cdot 012 = \cdot 051$ lb. per lb. air.

From Fig. 102 this agrees with the saturation temperature 106° F. Probably the air leaves non-saturated at a rather higher temperature than 106° F., but accepting this value,

$$\text{Mean difference of temperature} = 140 - \left(\frac{70+106}{2}\right) = 52^\circ \text{ F.}$$

$$\left.\begin{array}{l}\text{*Heat absorbed at evaporat-}\\ \text{ing surface per second}\end{array}\right\} = 192{\cdot}5 \times \frac{4}{5} \times \frac{1030}{1110} = 143 \text{ B.Th.U.}$$

$$\left.\begin{array}{l}\text{Heat absorbed per sq. ft. per}\\ \text{sec.}\end{array}\right\} = \frac{143}{248 \times \frac{13}{27}} = 1{\cdot}2 \text{ B.Th.U.}$$

$$\left.\begin{array}{l}\therefore \text{ Heat per sq. ft. per sec. per}\\ \text{deg. Fahr. diff. between}\\ \text{water and air}\end{array}\right\} = \frac{143}{248 \times \frac{13}{27} \times 52} = {\cdot}023 \text{ B.Th.U.}$$

In a similar calculation it was deduced that the corresponding rate of heat transmission at the evaporating surface for the lower pass was ·013. These values indicate that the rate of heat transmission increases rapidly with the temperature of evaporation.

According to the values deduced the rate of air-flow over the surface of the water was—

$$\frac{w}{a} = v\rho = 38 \times \frac{1}{14{\cdot}2} = 2{\cdot}7.$$

In the above paper Mr. Fitts also gave a curve showing the evaporation per square foot of surface at different water temperatures, both when working with the fan and when exposed only to still air. The full data, however, on which these curves were based were not given, but accepting his results the evaporation in still air was about one-sixth of that obtained when the fan was working at the rate mentioned in Table 22.

The experimental determination of the loss of heat from cooling ponds or reservoirs is somewhat difficult, and at the best only very approximate, because of the large masses of water involved and the uncertainties respecting the average temperatures. The state of the weather and the humidity of the atmosphere also influence the rate of evaporation to a considerable extent.

The following test was made by H. W. Barker† on a cooling

* The fraction $\frac{4}{5}$ allows for heat absorbed from underside of pans.

† "Cooling Reservoirs for Condensing Engines," *Proc. Inst. Civil Engs.*, Vol. CXXXII, 1897–98, Pt. II.

pond supplying condensing water for a triple-expansion engine working at about 1350 I.H.P. during the day, and also for a compound engine at about 300 I.H.P. during the night. There were two reservoirs. The water from the hotwell was carried in cast-iron troughs to the most distant one, from the bottom of which, at the opposite end, the coldest water was taken and delivered on to the surface of the second. The injection water was taken from the bottom of the other end of this second pond. The following results were observed or calculated :—

TABLE 23

Average temperature of air during week		61° F.
Temperature of injection water,	Monday morning	79° F.
	Saturday noon	88° F.
Temperature of hot-well,	Monday morning	117° F.
	Saturday noon	124° F.
Mean difference of temperature between water and air		41° F.
Heat dissipated per sq. ft. per sec. B.Th.U.		·08
,, ,, ,, ,,	per degree Fahr. difference water and air, B.Th.U.	·002

The results of several weeks' observations on the cooling of condensing water in a pond at Wampum, Pa., were given in a paper* by W. B. Ruggles. The area of the pond was about 288,000 sq. ft.; average depth 5·4 ft.; capacity 1,543,000 cub. ft. The water from a barometric condenser flowed about 50 ft. through a tile pipe, into the east end of the reservoir about 100 ft. from the north end. A dyke was built south of this inlet extending about 50 ft. towards the centre of the reservoir and then north nearly to the north end. This compelled the water to flow a distance of about 1100 ft. before it reached the intake to the power-house. Temperatures taken about 100 ft. apart showed that the mean water temperature was practically an average between the inlet and outlet. The make up water was pumped into the pond from a river. The following results given in Table 24 are based upon the observed temperatures and the calculated amount of heat and water dealt with during each week, allowing for the heating up of the make-up supply to the pond, and the

* *Trans. Amer. Soc. Mech. Engs.*, Vol. 34, 1912.

heat absorbed by the water in the pond during the period of each test.

TABLE 24

Date of test	Week ending 7th May, 1911.	Week ending 12th July, 1911.	Week ending 27th Nov., 1911.
Average temperature of air, ° F.	51·0	78·4	33·3
Average humidity of air, per cent	58·5	62·3	71·2
Average temperature of intake to power-house, ° F.	72·75	91·4	61·7
,, ,, ,, water from condenser, ° F.	101·4	129·4	90·7
,, ,, ,, reservoir, ° F.	87·0	110·0	76·7
,, ,, difference between water and air, °F.	36·0	31·6	43·4
Heat dissipated per sq. ft. per sec., B.Th.U.	·037	·033	·039
,, ,, ,, ,, ,, per degree Fahr. difference water and air B.Th.U.	·00102	·00104	·0009

An interesting and instructive series of simple experiments on the rate of evaporation of water at different temperatures placed in calm air were made by Thomas Box.* He suspended from a very delicate balance a vessel 12 in. square, 2¼ in. deep, containing about 7¾ lb. of hot water. To prevent losses from the sides and bottom of the vessel he enclosed it in another vessel with about an inch of wadding between. The atmosphere had a temperature 52° F. and humidity about 86%. During each experiment the temperature of the water and the weight evaporated (usually a small fraction of a pound) were noted and the time taken. From the known weight of water and the drop of temperature the amount of heat dissipated was easily calculated. Twenty-one experiments were made and the results plotted to smooth out the incidental experimental errors. The total heat dissipated per sq. ft. per second and per degree Fahr. difference of temperature are shown in Fig. 103. It will be noticed that curve A in this figure is similar in character to the total heat per lb. of air curve in Fig. 102, p. 216, and also similar to the curve of weight of vapour per lb. of saturated air. Another interesting point to notice is that the heat dissipated at the boiling

* *A Practical Treatise on Heat*

temperature (212° F.) is what would be expected from the general nature of curve A. The curve B in Fig. 103 signified that the rate at which heat is dissipated is much greater than might be expected from the difference of temperature between

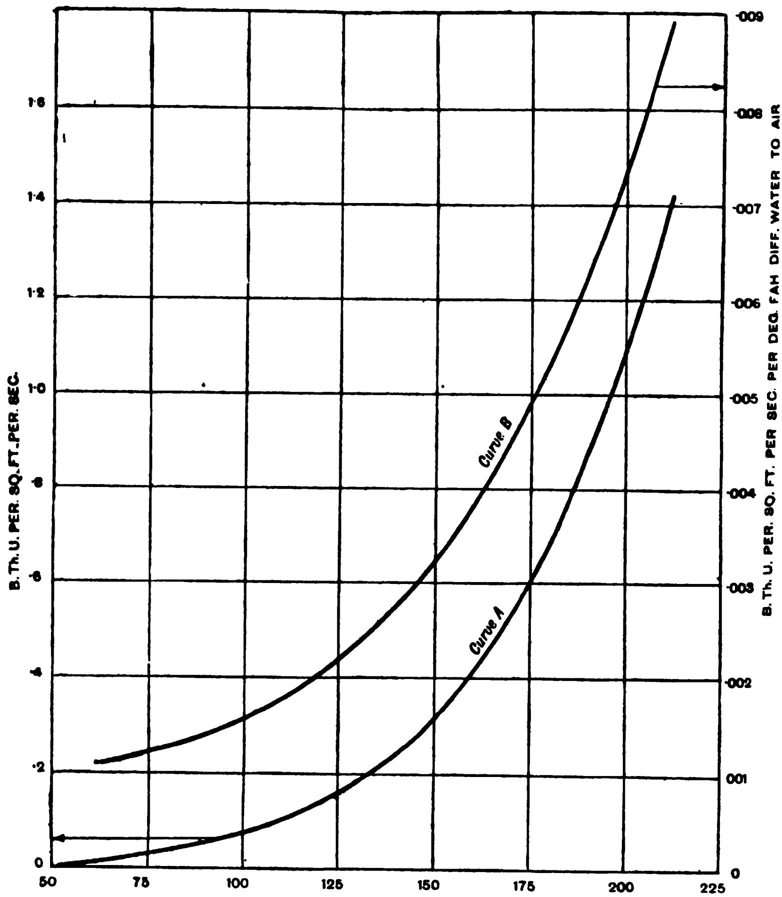

FIG. 103.—Loss of heat by evaporation from water in calm air; Box's experiments.

the water and the air in contact with it. In fact, Dalton's theory of vapour pressure states that the rate of evaporation is proportional to the difference between the vapour pressure corresponding to the temperature of the water and the vapour

pressure in the surrounding or enclosing atmosphere. It is obvious that if the water were completely enclosed in a vessel the pressure of the vapour in the enclosed air would ultimately equal the vapour pressure of the water, but when the vessel is open, currents are set up in the air removing the air which has become more or less saturated and replacing it with fresh air. It can be shown that saturated air is less dense than non-saturated air at the same total pressure, by calculations similar to those given on p. 102, and thus the air over the surface of water as it absorbs vapour tends to rise. Experience also shows that the rate of evaporation increases if the air velocity relative to the surface of the water increases either due to wind or due to mechanical draught.

Box tested the results of his experiments in the light of Dalton's theory of vapour pressure and found that they nearly obeyed the law.

$$E=(a+bt)\,(P-p) \quad \ldots\ldots\ldots\ldots \quad (1)$$

where E = evaporation per unit of area and time
t = temperature of water
P = vapour pressure at water temperature.
p = ,, ,, in surrounding air.

The writer has checked this relation by plotting the values of $\frac{E}{P-p}$ against the temperature t.

Expressing E in pounds of evaporation per sq. ft. per hour, t in degree Fahr., P and p in inches of mercury, it was found that between the water temperatures 100° F. and 212° F. the results were closely represented by—

$$E=(\cdot0313+\cdot000545t)\,(P-p) \quad \ldots\ldots\ldots \quad (2)$$

At about 80° F. this law gives values a little lower, and at 52° F. slightly higher than the experimental results.

For example, if $t=102°$ F., $P=2\cdot045$ in. mercury, p at 52° F. and humidity 86% is $\cdot3903\times\cdot86=\cdot335$ in. mercury.

Thus, $E=(\cdot0313+\cdot000545\times102)\,(2\cdot045-\cdot335)$
$=\cdot149$ lb. per sq. ft. per hour.

It would also be recognised from the above equation that when the temperature of the water is near that of the air the humidity of the air is an important factor because it affects the value of p. If the water has a high temperature then the

value of P is correspondingly large and the influence of p and the humidity of the atmosphere is relatively much less.

Box's experiments were made on a vessel of only 1 sq. ft. surface area. With cooling ponds of large area it is probable that the rate of cooling and the rate of evaporation in calm air would be somewhat less than in the above experiments because the currents of air would hardly be able to flow as readily across a large surface as with a small surface. The wind and rain also introduce other factors which are hardly calculable. In the case of running water in an open channel it is likely that the rate at which heat is dissipated to the atmosphere is greater than with practically stationary water.

Box also made a few experiments on the rate of evaporation from water at atmospheric temperature under different conditions of wind. He used a vessel $3\frac{1}{16}$ in. diameter suspended from a delicate balance. The results obtained are given below in Table 25 and it is seen that the wind had a considerable influence on the rate of evaporation.

TABLE 25

Liquid.	State of wind.	Temp. of air. °F.	Humidity of air, per cent.	Pounds evaporated per sq. ft. per hour.	Calculated evaporation for still air. Lbs. sq. ft. per hour.	Ratio. $\frac{\text{Evaporation with wind.}}{\text{Evaporation in calm air.}}$
Water .	Dead calm .	60	71	·0081	·0081	—
do. .	Fresh breeze	57	57	·0475	·0109	4·4
do. .	Strong wind	53	59	·0781	·0089	8·8
do. .	Gale . .	52	74	·0672	·0054	12·4
1	2	3	4	5	6	7

From his experiments Box also deduced for the evaporation in still air with the water at or near atmospheric temperature,

Evaporation $=(\mathrm{P}-p)\cdot 054$ lb. per sq. ft. per hour.

where P = vapour pressure of water at the temperature of the air, inches mercury.

p = vapour pressure of the air as shown by the hygrometer, inches mercury.

This expression has been used to give column 6 in Table 25, and column 7 has been derived by dividing column 5 by the

values in column 6. From these results it is seen that the velocity of the air has an important influence on the rate of evaporation.

Box also quotes the results of some experiments made over lengthy periods with water at natural temperatures. In one case a Mr. Greaves made a light slate cistern and set it afloat on a raft in a mill-stream. It was 3 ft. square, 1 ft. deep, and immersed to a depth of 4 in., with 4 to 6 in. of water continually in it. This cistern was fully exposed to the sun, wind, rain, and evaporation, and the depth of the water in it was measured periodically over ten years, being compared with an adjacent rain gauge. The mean result showed an average evaporation of 21 in. per annum, the mean rainfall for the same period being 25½ in. Mr. Fletcher found an annual evaporation of 47½ in., as a mean of seven years' observations with a small vessel fully exposed 5 ft. above the ground, and a Mr. Miller at Whitehaven obtained an annual evaporation of 30 in. as a mean of six years when the mean rainfall was 45·25 in. In this case, the mean evaporation varied from ·88 in. in the month of January to 4·547 in. in June, decreasing again to 1·087 in. for the month of December. These various results therefore indicate that, even at natural temperatures, the natural rate of evaporation is not far from the amount of rainfall on the same surface in this country, and therefore it seems obvious that, considering the much greater rate of evaporation in ponds for cooling condensing water, provision is generally needed for a make-up supply, either in the way of fresh feed for the boilers or for a direct supply to the pond where the feed water is taken from the pond or reservoir, not only to balance the loss by evaporation but also to make up for leakage.

Whilst the required size of cooling pond depends upon the amount of heat to be dissipated it depends also on the character of the load on the steam power plant. If a plant works continuously night and day the cooling pond would need to be of comparatively large area, but could be made shallow if the make up supply was certain in periods of drought. In the case of a plant working on a day load only, and particularly if stopped at week-ends, the pond may have a much smaller

area than in the other case, but may with advantage be deeper, so that during the working period there would be a comparatively large mass of water heating up slowly, and cooling again during the night and at week-ends.

The following calculations will illustrate how the dimensions of ordinary cooling ponds may be calculated approximately.

EXAMPLE I.—A steam engine working night and day at 1000 I.H.P. uses 13 lb. of dry saturated steam per I.H.P. per hour. The steam pressure is 170 lb. sq. in. absolute and mean exhaust temperature 120° F. The mean atmospheric temperature is 52° F., and the water is to be cooled from 120° F. to 80° F. in the pond.

From steam tables :—

Total heat supplied to engine per lb. steam $=1195-87{\cdot}9$
$=1107$ B.Th.U.

Heat converted to work per lb. steam $=\dfrac{1\times 33000\times 60}{778\times 13}$
$=196$ B.Th.U.

Loss by radiation, etc., from engine, $1107\times{\cdot}05=55$B.Th.U.,say.

Heat sent to condenser per lb. steam $=1107-196-55$
$=856$ B.Th.U.

Heat per I.H.P. per second to pond $=\dfrac{856\times 13}{3600}$
$=3{\cdot}09$ B.Th.U.

Heat dissipated per sq. ft. per sec. at 80° F. $={\cdot}04$ B.Th.U., say

Heat dissipated per sq. ft. per sec. at 120° F. $={\cdot}13$ B.Th.U., say

Mean $={\cdot}085$ B.Th.U.

Mean water temperature $\dfrac{80+120}{2}=100°$ F., say.

Heat dissipated per sq. ft. per sec. at 100° F. $={\cdot}07$ B.Th.U.

Probable mean $=\dfrac{{\cdot}07+{\cdot}085}{2}$
$={\cdot}078$ B.Th.U.

$\therefore$ Area of surface per I.H.P. $=\dfrac{3{\cdot}09}{{\cdot}078}=39{\cdot}6$, say, 40 sq. ft.,

or, for 1000 I.H.P. $=40{,}000$ sq. ft.

The depth of the pond or reservoir would depend upon whether cheap make-up water were available or not. For example, if it were desirable to provide a sufficient volume of water so as to be able to dispense with make-up supply for one month, say, then,

Probable mean rate of evaporation per sq. ft. per hour $= \cdot 15$ lb., or, for one month of four weeks the evaporation per I.H.P. would be

$$\cdot 15 \times 24 \times 7 \times 4 \times 40 = 4030 \text{ lb.}$$

$$\text{or, } = \frac{4030}{62} = 65 \text{ cub. ft.}$$

$$\therefore \quad 40 \times \text{depth} = 65$$

$$\text{or, depth} = \frac{65}{40} = 1 \cdot 63 \text{ ft.}$$

or, say, 2 ft. roughly.

In addition to this, some provision would need to be made for accumulation of mud and sand at the bottom of the pond, say, 2 ft., and thus the total depth might be made 4 ft. for the normal water level.

Example II.—The same engine as in Example I working night and day except for a week-end stoppage of thirty hours. During the week-end it is required to allow the pond to cool through an average of 20° F. in the thirty hours. The same quantity of cooling water is supplied per working hour as in the previous example and the mean pond temperature is again 100° F.

Thus the outlet and inlet temperatures at the pond are: Soon after starting on Monday, 70° F. and 110° F. respectively, and on stopping for the week-end are, 90° F. and 130° F. respectively.

Working as in the previous example the mean rate at which heat is dissipated per sq. ft. of surface per second is, on Monday morning, ·058, and on stopping ·108, giving a mean of $\frac{\cdot 058 + \cdot 108}{2}$ $= \cdot 083$ for the working period. During the week-end stoppage the same value ·083 would represent sufficiently nearly the rate of heat loss per sq. ft. per second.

Heat supplied to pond in 138 hours per I.H.P.

$$=3{\cdot}09\times3600\times138$$
$$=1{,}535{,}000 \text{ B.Th.U.}$$

This is divided into two parts, the loss from the pond during working hours and the loss during the stoppage. Thus, during working 138 hours heat lost per I.H.P.

$$=1{,}535{,}000\times\frac{138}{(138+30)}$$
$$=1{,}260{,}000 \text{ B.Th.U.}$$

$$\therefore \text{ Area of pond per I.H.P.}=\frac{1{,}260{,}000}{{\cdot}083\times3600\times138}=30{\cdot}6 \text{ sq. ft.}$$

or, $=31$ sq. ft., say,

and for 1000 I.H.P., total area $=31{,}000$ sq. ft.

The mass of water in the pond has to be sufficient to store $1{,}535{,}000-1{,}260{,}000=275{,}000$ B.Th.U. per I.H.P. for dissipation during the week-end period of thirty hours, and since the water cools 20° F. on the average,

$$\text{Then, Weight of water}=\frac{275{,}000}{20}=13{,}750 \text{ lb.}$$

$$\text{Volume}=\frac{13{,}750}{62}=222 \text{ cub. ft.}$$

Thus, $31\times\text{depth}=222$

$$\text{or, Depth}=\frac{222}{31}=7{\cdot}16 \text{ ft.}$$

or, say, 7 ft.

Here again two or three feet might be added to the depth on account of accumulation of sand and mud at the bottom, and to allow for the possible lack of make-up supply in case of drought.

CHAPTER III

TRANSMISSION OF HEAT IN EVAPORATORS

Evaporators.—Evaporation is commonly applied to the concentration of liquid solutions. This may be accomplished by boiling the liquid over a fire or furnace, but more usually in practice the heat for evaporation is obtained from condensing steam, where the steam is allowed to condense on the surface of tubes and evaporation proceeds at the other surface at a lower temperature. Not only is it convenient to utilise steam in this way, but considerable economy in evaporation may be obtained by making use of multiple effect evaporation. In this arrangement steam is supplied to the first effect of the series wherein it is condensed either inside or outside suitably arranged tubes. At the other surface the heat supplied evaporates some of the water from the boiling liquid, and the vapour or steam thus formed is then led to the second effect of the series in which the same process goes on as in the first but at a lower temperature, with the result that more than one pound of water can be evaporated per pound of steam supplied to the first effect. For example, with a six-effect evaporator the evaporation is something like four or five times the amount of steam used.

Special considerations sometimes prevents the adoption of such an elaborate arrangement of multiple-effect evaporator, as, for instance, where the nature of the liquid makes it undesirable to raise it to a comparatively high temperature, and only one or two stages are then used. Or the single stage evaporator may be used when the rate of heat transmission is low and when close regulation is necessary, as in " vacuum pans " for the final concentration of sugar solutions. Also the evaporators used on board ship for obtaining distilled water

from sea water for the make-up supply and other purposes are commonly made single stage because considerations of weight and space prevents the use of an elaborate arrangement. But in such cases reasonable economy may be obtained even by using only single evaporators, taking steam either from the boilers or from some lower pressure stage at the engines, and then sending the steam generated at the evaporator into another lower pressure stage or to the feed water heater. Whilst somewhat less than a pound of water is thereby evaporated per pound of steam condensed in the evaporator, a certain amount of work may also be obtained at the engines

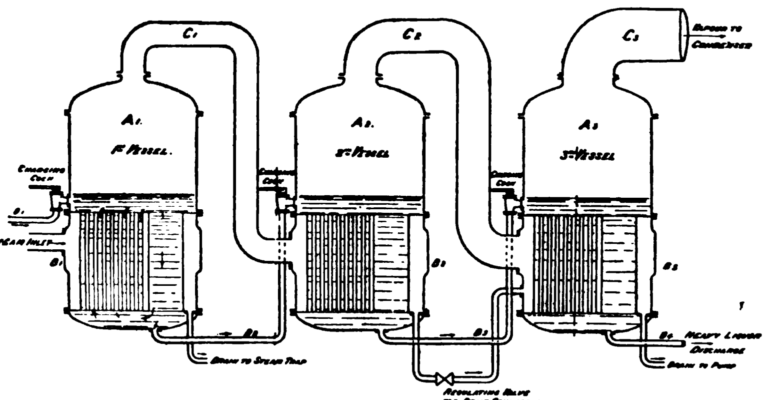

FIG. 104.—Triple effect evaporator of the calandria type.

by the procedure above mentioned, either from the steam before it reaches the evaporator, or from the steam generated in the evaporator, or in some cases from both. If the steam generated in the evaporator is sent to the feed heater most of the heat in the steam is returned to the feed water.

Multiple-Effect Evaporators.—These evaporators are used for the concentration of sugar solutions, meat extracts, glue, gelatine, caustic soda and other chemical solutions, waste alkaline liquors from paper mills, wood extracts for tanning or dyeing, for the production of fresh water from sea water, etc.

The diagrammatic sketch in Fig. 104 illustrates a three-effect evaporator of the calandria type. The liquid boils inside the tubes and the steam condenses on the outside, a large down-

comer tube being also introduced to facilitate the circulation of the liquid. The vapour or steam formed in effect I passes to the heating side of effect II, and similarly the vapour formed in II is the heating steam for effect III. The steam supplied to effect I may be taken directly from the boilers, or, as is a common and economical practice, the exhaust steam from the mill engines at about 5 lb. per sq. in. gauge is utilised for the purpose. The water of condensation from No. 1 effect is usually returned as hot feed to the boilers.

The hot liquid to be concentrated passes into effect I. Due to evaporation a certain amount of concentration takes place and the liquid passes over to effect II, where further

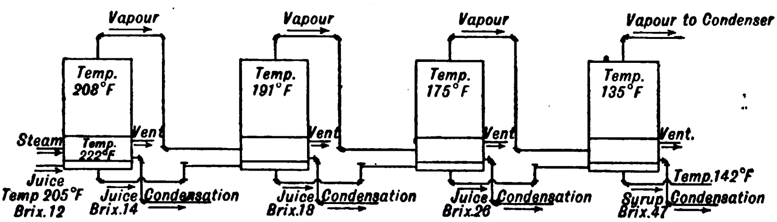

FIG. 105.—Temperatures and densities in quadruple-effect sugar-juice evaporator.

concentration is obtained. Similarly, the final concentration of the liquid takes place in No. III effect, and from this the concentrated solution is extracted by a pump.

One of the most important uses of multiple evaporators is the concentration of sugar solutions. In a sugar factory the process of evaporation consists of two parts: (1) The juice is supplied to a multiple evaporator with a density of 12 to 18 degree Brix. and is concentrated to about 55 degree Brix. (2) The syrup is then further concentrated in a single-effect vacuum pan to a final density of about 95 degree Brix. under conditions suitable for crystallising the sugar. The multiple evaporator may have two, three or four effects according to circumstances, the first effect being supplied with steam at a little above atmospheric pressure; commonly the exhaust steam from the factory engines is used for this purpose. Fig. 105 illustrates diagrammatically the sequence and arrangement of a quadruple-effect evaporator with temperatures and densities

obtained from a test under factory conditions.* The juice is supplied to the first body or effect and then passes in succession to the other effects. The density of the juice leaving the last effect is regulated by hand valves placed between each pair of effects on the liquor pipes.

Vacuum pans, as commonly constructed, consist of a vessel or calandria containing one or more conical copper coils. The heating steam is supplied to the inside of the coil, and the vapour from the vessel passes to a condenser.

The calandria type of evaporator, shown in Fig. 104, is particularly suitable for dealing with solutions of a frothy character, since it allows large cubic capacity and little or no liquid is carried over by the steam into the next evaporator. A disadvantage under some circumstances is the large bulk and weight for the amount of evaporation obtained. Also, when concentrating a liquid such as glue or gelatine it is desirable to have the concentration carried out in the shortest possible time of contact as heat treatment of long duration tends to cause a darkening of the resultant product. A six-effect multiple evaporator† of compact arrangement is shown diagrammatically in Fig. 106, representing the Yaryan evaporator as made by Mirrlees Watson Company, Ltd., of Glasgow. The effects are numbered 1 to 6 in order, effect 1 working under relatively high temperature conditions, the temperature falling between each stage until effect 6 is reached, the conditions being then near to ordinary condenser temperatures. This arrangement is also largely used for the distillation of water from sea water in arid countries where fresh water is not otherwise obtainable.

When evaporating sea water the steam generated in the last effect 6 is passed through two separators to ensure that no brine passes with the vapour into the 1st heater and into the condenser. Part of the sea water, after circulating through the condenser, is taken into the 1st heater, which is a tubular condenser of special form, and this water is heated at the expense of the latent heat in the steam from the last effect.

* E. W. Kerr, "Capacity and Economy of Multiple Evaporators," *Jour. Amer. Soc. Mech. Engs.*, July, 1916.

† Reproduced from paper on "Multiple Effect Evaporation," by Mr. C. Day, *Manch. Ass. Engs.*, 1905.

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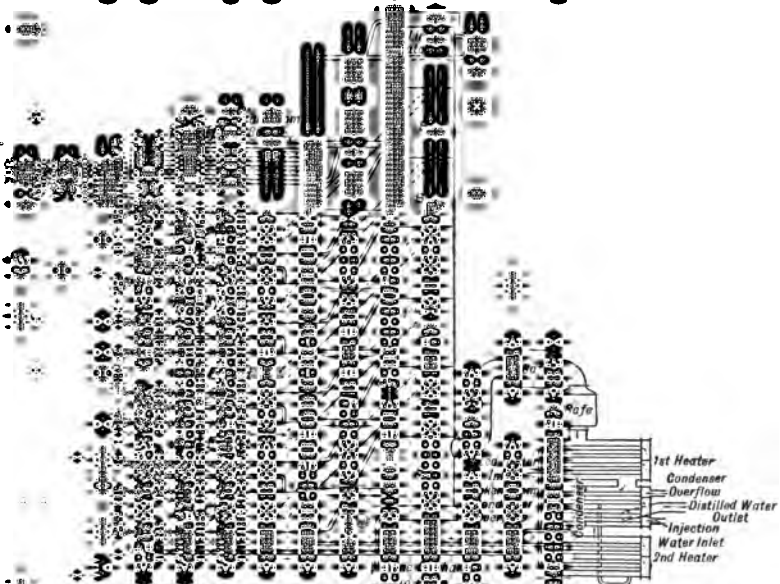

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The heating steam in the 1st effect is supplied directly from the boilers through a throttling valve at about 40 lb. per sq. in. gauge, and the exhaust steam from the various steam pumps on the plant also exhaust at this pressure into this effect. The water of condensation from this effect is usually returned to the boilers. The circulating brine and the steam generated in this 1st effect pass through the special separator S_1, and the steam from this separator is passed into

TABLE 26

Number of effect or stage.	Approximate steam pressure lbs. sq. in. absolute.	Approximate steam temperature °F.	Approximate difference of temperature (steam to vapour) at each stage, °F.
1	56	288	26
2	36	262	26
3	23	236	26
4	14	210	28
5	8	182	30
6	4	152	36
Condenser	1·5	116	

the 2nd effect to evaporate more water from the brine led from the separator S_1 into the evaporation tubes in this effect. At each subsequent stage or effect the steam and brine are separated in a similar manner. The brine is gradually concentrated and is eventually extracted from the separator at the last evaporator 6 and discharged to waste by a special pump. The whole of the water of condensation obtained from the brine is eventually withdrawn by the vacuum pump from the condenser. It is seen from the illustration, Fig. 106, that the total water of condensation produced up to any stage or effect is led to the steam side of the next effect.

Pipe connections to the condenser vent the air from the

various effects, the cocks usually being left open just sufficiently for this purpose.

The action and purpose of multiple-effect evaporators will perhaps be more clearly understood from the following calculations relating to a six-effect evaporator for the production of distilled water from impure water or from sea water. For this

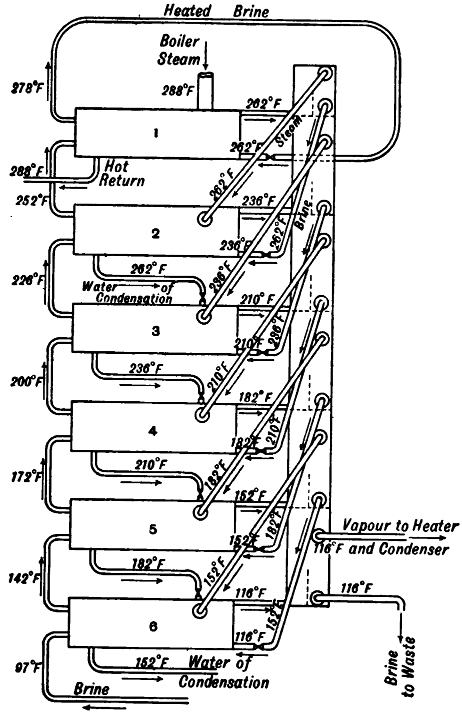

FIG. 107.—Approximate temperatures in sextuple effect evaporator when evaporating sea-water.

purpose take the steam pressures and temperatures at the various effects or stages 1 to 6 to be approximately represented by the values in Table 26.

For the purposes of comparison consider the following to represent four different conditions of operation.

Case (a)—With the temperature conditions specified in Table 26, and using the special heater tubes for the brine at

each effect, as in the Yaryan evaporator, so that the brine is heated to within 10° F. of the temperature of the heating steam in each effect. Brine enters the heater tubes of effect No. 6 at 97° F. The scheme of temperatures is exhibited by the sketch in Fig. 107.

Case (b)—Conditions as in (a) except that no heater tubes are used, the brine entering effect No. 1 at 97° F.

Case (c)—Conditions as in (a) but in addition the brine is heated by boiler steam in the special "lime catcher" from 278° F. to 310° F., and is then passed through a throttling valve into the evaporation tubes of effect No. 1 at 262° F.

Case (d)—Conditions as in (b) but in addition the brine is heated by boiler steam from 97° F. to 310° F. in the special "lime catcher" and is then passed through a throttle valve into the evaporation tubes of No. 1 effect at 262° F.

In the following calculations let it be further specified that 1 lb. of dry saturated steam at 288° F. is supplied to the No. 1 effect and that the water of condensation is returned to the boiler at 288° F., and that the water of condensation leaves each effect at the same temperature as the steam in that effect; also, that the concentrated brine is discharged to waste from effect 6 when from each pound of brine originally supplied three-quarters of a pound of water has been evaporated.

For calculations such as the following representing a first approximation it will be sufficiently accurate to take the specific heat of the brine as unity.

Let $y = \begin{cases} \text{pounds of brine supplied to evaporator per pound} \\ \text{of steam condensed in No. 1 effect.} \end{cases}$

Case (a)—Effect No. 1.

Latent heat available from 1 lb. steam at 288° F. = 920 B.Th.U.

„ „ of 1 lb. steam at 262° F. = 937 B.Th.U.

$\left.\begin{array}{l}\text{Heat required in heater tubes to} \\ \text{raise } y \text{ lb. brine through } 26° \text{ F.}\end{array}\right\} = y \times 26$ B.Th.U.

$$\text{Evaporation} = \frac{920 - y \times 26}{937} = (\cdot 982 - \cdot 0278y) \text{ lb.}$$

Let x lb. be the evaporation produced in throttling y lb. from 278 to 262° F.

Then, $y\,(278-262)=x\times 937$

$$x=\frac{16y}{937}=\cdot 0171y \text{ lb.}$$

Total evaporation $=\cdot 982-\cdot 0278y+\cdot 0171y$
$=(\cdot 982-\cdot 0107y)$ lb.

Weight of brine passing to next effect $=y-(\cdot 982-\cdot 0107y)$
$=(1\cdot 0107y-\cdot 982)$ lb.

Water of condensation drained to boiler $=1$ lb.

Effect No. 2.

Weight of steam from 1 $=(\cdot 982-\cdot 0107y)$ lb.

Heat available by condensation $=(\cdot 982-\cdot 0107y)\times 937$
$=(921-10\cdot 03y)$ B.Th.U

Heat to raise y lb. brine through 26° F. $=y\times 26$ B.Th.U.

Evaporation $=\dfrac{921-10\cdot 03y-26y}{955}$
$=(\cdot 965-\cdot 0377y)$ lb.

Evaporation by throttling of brine,

$$(1\cdot 0107y-\cdot 982)\times 26=x\times 955$$
$$x=(\cdot 02755y-\cdot 02675)\text{ lb.}$$

Total evaporation $=(\cdot 965-\cdot 0377y)+(\cdot 02755y-\cdot 02675)$
$=(\cdot 9382-\cdot 0101y)$ lb.

Brine to No. 3 effect $=(1\cdot 0107y-\cdot 982)-(\cdot 9382-\cdot 0101y)$
$=(1\cdot 0208y-1\cdot 9202)$ lb.

Drainage to No. 3 effect $=(\cdot 982-\cdot 0107y)$ lb.

Calculating in this manner at each stage we get finally

Effect No. 6.

Heat available by condensation $=(\cdot 805-\cdot 00862y)\times 1006$
$=(810-8\cdot 67y)$ B.Th.U. (1)

Heat available by throttling drainage through 30° F. $=(3\cdot 6688-\cdot 0395y)30$
$=(110-1\cdot 185y)$ B.Th.U. (2)

Heat available by throttling brine through 36° F. $=(1\cdot 0481y-4\cdot 4738)36$
$=(37\cdot 7y-161)$ B.Th.U. (3)

Heat for y lb. brine $=y\times(142-97)=y\times 45$ B.Th.U. (4)

Total evaporation in 6 $= \dfrac{(1)+(2)+(3)-(4)}{1027}$

$= (\cdot739 - \cdot0167y)$ lb.

Brine to overflow $= (1\cdot0481y - 4\cdot4738) - (\cdot739 - \cdot0167y)$

$= (1\cdot0648y - 5\cdot2128)$ lb.

Drainage to condenser $= (3\cdot6688 - \cdot0395y + \cdot805 - \cdot00862y)$

$= (4\cdot4738 - \cdot0481y)$ lb.

But, $1\cdot0648y - 5\cdot2128 = \cdot25y$ (specified on p. 246)

or, $\cdot8148y = 5\cdot2128$

or, $y = 6\cdot4$ lb.

Water evaporated $= 6\cdot4 \times \cdot75 = 4\cdot8$ lb.

That is, 4·8 lb. of water have been evaporated per pound of boiler steam condensed in the effect No. 1.

Allowing, say, a 10% reduction of evaporation due to external radiation, etc., the evaporation then becomes $4\cdot8 \times \cdot9 = 4\cdot3$ lb.

Also, probable brine supply per lb. steam $= 6\cdot4 \times \cdot9 = 5\cdot8$ lb.

Assuming the boiler could generate 11 lb. of steam per pound of coal burned under the ordinary conditions of operation, then the total evaporation obtained per pound of coal burned becomes $4\cdot3 \times 11 = 47\cdot3$ lb.

Neglecting all losses of heat by radiation, etc., it can be shown that the total heat rejected by the plant must be equal to that supplied by the heating steam in the No. 1 effect. Thus, if $t°$ F. is the sea-water temperature, and taking the specific heat to be unity,

Heat lost by brine at discharge $= \cdot25 \times 6\cdot4 \times (116 - t)$ B.Th.U. (1)

Heat rejected by No. 6 effect to heaters and condensers $\Big\} = (4\cdot4738 - \cdot0481y)(152 - t) + (\cdot739 - \cdot0167y)(1027 + 116 - t)$

$= 4\cdot166(152 - t) + \cdot632(1143 - t)$ B.Th.U. (2)

Heat recovered by brine at heaters $\Big\} = 6\cdot4(97 - t)$ B.Th.U. (3)

Heat rejected by plant $=(1)+(2)-(3)$

$$=(185-1{\cdot}6t)+(633-4{\cdot}166t)+(723-{\cdot}632t)-(621-6{\cdot}4t)$$
$$=920 \text{ B.Th.U.}$$

This is the amount of heat supplied by the 1-lb. heating steam in the No. 1 effect.

It also follows that even had the calculations taken into account certain radiation losses occurring at each effect, the heat supplied by the heating steam at No. 1 effect would be equal to the heat subsequently lost externally by radiation plus that rejected by the condenser circulating water and the brine discharge.

Neglecting the external losses of heat the amount of water evaporated in the several effects and the amount of brine entering are as follows :—

Number of effect.	Water evaporated per lb. steam supply, lbs.	Weight of brine entering each effect per lb. steam supply.	
		* Evaporation tubes, lbs.	Heater tubes, lbs.
1	·914	6·4	6·4
2	·874	5·49	6·4
3	·835	4·61	6·4
4	·794	3·78	6·4
5	·750	2·98	6·4
6	·633	2·23	6·4
	Total 4·8 lb.	1·6 to overflow.	—

Case (b)—Brine heated in No. 1 effect from 97° F. to 262° F. Calculations similar to those for conditions (a) lead to the following results without allowing for external losses of heat :—

$y=4{\cdot}05$ lb.

${\cdot}75y=3{\cdot}04$ lb.

$=$Total evaporation per pound steam used in No. 1 effect.

Again deducting 10% as an allowance for the reduction of evaporation due to external heat losses, then,

Total evaporation $=3{\cdot}04\times{\cdot}9=2{\cdot}74$ lb.

* The evaporation produced by the throttling of the brine supply at each effect is neglected in this tabulation.

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Similar calculations may be made with an allowance for heat losses at each separate effect. It would be found that it is more important to prevent external losses of heat at the high temperature end of the evaporator than at the low temperature end. The reason for this is that losses of heat at No. 1 effect, for example, represent a corresponding loss of evaporation at the whole of the succeeding effects, whereas loss at No. 6 effect, say, has already given its quota of evaporation at the preceding effects.

Experimental Data from Evaporators.—The resistance offered to the flow of heat from the heating steam to the boiling liquid is due to (1) The resistance between the steam and the metal surface, (2) The resistance of the metal of the tubes, and (3) The resistance between the metal and the boiling liquid.

As in surface condensers the resistance between the heating steam and the tube depends upon—

(a) The temperature, velocity, and the manner of distribution of the heating steam among the tubes.

(b) The presence of incondensable gases (air) in the steam and the arrangements made for its withdrawal, and water of condensation on the tubes.

(c) The cleanliness of the tubes (steam side).

The resistance of the metal depends upon its conductivity and thickness.

The resistance between the tubes and the liquid depends upon—

(d) Velocity of circulation of the liquid.

(e) Character, viscosity, and density of the liquid.

(f) Cleanliness of the tubes (liquid side).

It is obvious that the amount of heat transmitted through unit surface in unit time is inversely proportional to the total resistance and practically proportional to the difference of temperature, all other things equal. The difference of temperature between the heating steam and the boiling liquid depends upon—

(g) Pressure of the heating steam.

(h) Pressure of the vapour formed,

(i) Partial pressure of the air in the heating steam.
(j) Hydrostatic head of the boiling liquid.
(k) Character and density of the boiling liquid.

The hydrostatic head or depth of submergence necessarily raises the boiling temperature of the liquid above that of the vapour over the liquid, and therefore the difference of temperature between the heating steam and the boiling liquid is less than between the steam and the vapour, causing a proportionate reduction in the amount of heat transmitted. This influence is greatest at the lowest boiling temperature (i.e. the last effect), as is shown by the values on p. 274. The presence of solids in solution in water also raises the boiling temperature for any given vapour pressure over the liquid, and the more concentrated the solution becomes the greater is the influence on the boiling temperature. These two causes together partly account for the greater difference of temperature required between the heating steam and the vapour formed as the concentration proceeds and as the pressure falls from body to body in multiple evaporators.

It is seen then that the problem of heat transmission in multiple evaporators is very complex, in fact it is practically impossible to ascertain the exact influence of all the separate factors affecting the transmission of heat. So far as the conditions on the steam side of the tubes are concerned, however, the problem is much the same as in ordinary surface condensers, but on the liquid side of the tubes the conditions are much more complicated in multiple evaporators than in ordinary surface condensers. For example, the rate of circulation of the liquid is practically indeterminate, being influenced not only by the arrangement of the apparatus and the rate of evaporation, but also by the increasing viscosity of the liquid as concentration proceeds and as the temperature falls from body to body. Also the surface of the tubes is commonly subject to a progressive fouling as time goes on until cleaning becomes necessary.

With the experimental results available it is only possible, then, to discuss in detail some of the various factors mentioned on p. 251. Considering first the influence of the conductivity and thickness of the metal tubes, it is instructive to compare

the relative merits of copper, brass, and mild steel tubes so far as the transmission of heat is affected. An example will illustrate the method of comparison :—

Heat transmitted per sq. ft. per second $=2{\cdot}4$ B.Th.U.

Copper tubes	·065 in. thick,	conductivity ·06, say.	ft. lb. sec. units.
Brass tubes	·065 ,, ,,	,, ·018 ,,	
Mild steel tubes	·12 ,, ,,	,, ·007 ,,	

Temperature difference between heating steam and vapour when using brass tubes 20° F.

If $\theta_1-\theta_2$ is the fall of temperature through the metal,

Then, for copper tube, $\theta_1-\theta_2=\dfrac{2{\cdot}4\times{\cdot}065}{12\times{\cdot}06}={\cdot}22°$ F.

,, brass ,, $\theta_1-\theta_2=\dfrac{2{\cdot}4\times{\cdot}065}{12\times{\cdot}018}={\cdot}72°$ F.

,, steel ,, $\theta_1-\theta_2=\dfrac{2{\cdot}4\times{\cdot}12}{12\times{\cdot}007}=3{\cdot}4°$ F.

Taking it that the resistances between the steam and the metal and between the metal and the liquid are the same for all these metals, then,

$$\left(\begin{array}{l}\text{Fall of tempera-}\\ \text{ture (steam}\\ \text{to vapour)}\end{array}\right)-\left(\begin{array}{l}\text{Fall of tempera-}\\ \text{ture through}\\ \text{metal}\end{array}\right)=20-{\cdot}72=19{\cdot}28°\text{ F.}$$

Thus, Fall of temperature from steam to vapour

$=19{\cdot}28+{\cdot}22=19{\cdot}5°$ F. for copper tube.

$=19{\cdot}28+{\cdot}72=20°$ F. for brass tube.

$=19{\cdot}28+3{\cdot}4=22{\cdot}7°$ F. for steel tube.

Or, to state the results in another way, with the available fall of temperature from steam to vapour 20° F. in all cases, and with the heat transmitted by unit surface of the brass tubes represented by unity,

Then, Heat transmitted $=\dfrac{20-{\cdot}22}{20-{\cdot}72}=1{\cdot}025$ for copper tube.

$=\dfrac{20-{\cdot}72}{20-{\cdot}72}=1$ for brass tube.

$=\dfrac{20-3{\cdot}4}{20-{\cdot}72}={\cdot}86$ for mild steel tube.

Influence of Circulation of Boiling Liquid on the Rate of Heat Transmission between the Metal Surface and the Liquid. —Austin's experiments discussed previously showed definitely that violent stirring of the boiling liquid had an appreciable influence in reducing the difference of temperature between the surface and the liquid for any given transmission of heat. The writer has reproduced the results shown in Figs. 100, 101, and 98 of *Heat Transmission by Radiation, Conduction, and Convection* in a modified form. In Fig. 108 the calculated rate of heat transmission between a metal surface and water has been plotted on a base of difference of temperature for Austin's experiments, and the numbers marked against the lines refer to the heat transmission in B.Th.U. per square foot per second. Comparing the curves A and B it is seen that with water boiling at 100° C. in contact with an iron surface the stirring of the water increased the rate of heat transmission to a considerable extent, and particularly so with small differences of temperature; but whilst for curve A the rate of heat transmission was practically independent of the temperature difference, with curve B, where the water was not stirred, the rate of heat transmission increased fairly rapidly with increases in the difference of temperature, that is with the rate of evaporation. The curves C, D, and E, illustrate the character of the results obtained with water at 20° C., 30° C. and 50° C. when stirred rapidly, but not boiling, in all these cases showing an improvement over water boiling at 100° C. (212° F.) when not stirred. The relations between curves C, D, E, and A, show to some extent the influence of the temperature of the water on the rate of heat transmission.

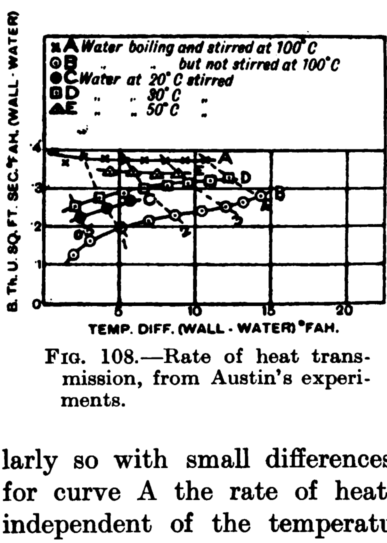

FIG. 108.—Rate of heat transmission, from Austin's experiments.

The results shown plotted in Fig. 109 were derived from

Fig. 98, p. 212, of *Heat Transmission by Radiation, Conduction, and Convection,* with reference to Bryant's experiments on the temperature of a metal surface in contact with boiling water. It is doubtful, however, whether the upper portion of this curve B in Fig. 109 truly represents the influence of the circulation of the water at the high rates of evaporation obtained.

Experiments of a General Nature.—Some experiments have been made by C. Lang* on one type of Weir single-effect evaporator as used on board ship at the date of the experiments. The evaporator, though not now of modern construction, was of the ordinary type stated to make ten tons of fresh water per day of twenty-four hours. The shell consisted of a steel cylindrical vessel 3 ft. diameter and 4 ft. 3 in. long. The heating surface was composed of twelve solid-drawn copper tubes, $1\frac{1}{2}$ in. external diameter, No. 10 B.W.G., giving a total heating surface of 38 sq. ft. These were U-shaped and fixed at both ends in the tube plate so that the tubes were horizontal. The steam condensed in the tubes and the brine was in contact with the outside surface.

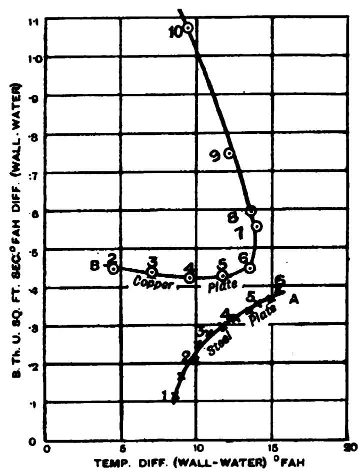

FIG. 109.—Rates of heat transmission, from Bryant's experiments.

The outlet ends of all the tubes except the lowest ones were contracted by screwing a brass plug into each of them having a small hole through it. The object of this was to increase the resistance to the flow of the steam through the tubes so as to make certain that none of the tubes would remain filled with stagnant air, and thus it ensured a definite flow through all the tubes. Steam was led to the evaporator from a boiler and a separator was placed between them to ensure practically dry steam. Salt water, having a density of 19 oz. of salt to the

* *Trans. Engs. and Shipbds. in Scotland,* Vol. XXXII.

gallon, was used in the evaporator. The steam evaporated from the salt water could be led either to a condenser or blown direct to the atmosphere. When the condenser was being used the water of condensation was tested from time to time to see that no salt water was carried over by the steam, the silver nitrate test being used for this purpose. The discharge of water of condensation from the evaporator was weighed and the heat transmission was calculated from this weight and the known amount of heat given up by each pound of steam. Presumably the evaporator was working with the tubes comparatively free from scale deposit.

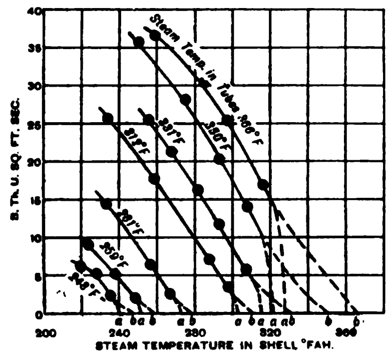

FIG. 110.—Relations between temperatures of heating steam and vapour, and the heat transmitted, from Lang's evaporator experiments.

The results obtained are shown graphically in Figs. 110 to 112. In Fig. 110 the heat-flow in B.Th.U. per sq. ft. of tube surface per second is plotted on the base of vapour temperature in the shell for the various temperatures of the heating steam indicated on the curves. The points marked *b* are at the same temperatures as the heating steam, and it is presumed that under these conditions of vapour temperature the flow of heat would be zero. The points on the zero line marked *a* were derived from the dotted curve AB in Fig. 111. In this figure the temperature of the water of condensation from the tubes for the various steam temperatures are plotted on the base of

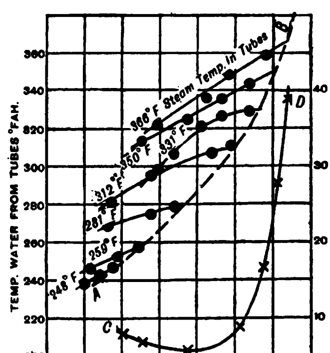

FIG. 111.—Relations between temperatures of heating steam, vapour, and the water of condensation, from Lang's evaporator experiments.

vapour temperatures. The dotted line AB is drawn through the points on the full line curves which coincide with the heating steam temperatures, and the corresponding vapour temperatures have been transferred to the points *a* in Fig. 110. The calculated difference between the steam and the vapour temperatures at these limiting conditions have been plotted in Fig. 111 as the line CD, and it will be noticed that, whilst the difference appears to be practically constant up to about the vapour temperature of 300° F., above this the difference increases very rapidly.

The curves in Fig. 110 have been continued to the points *a* and *b* on the zero line. It is seen that in no case does the

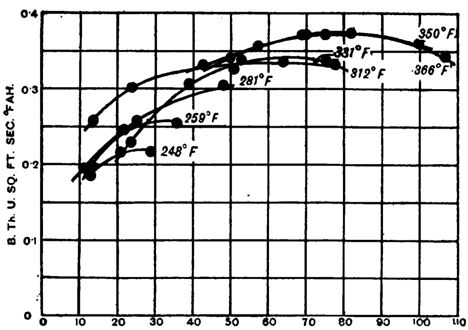

FIG. 112.—Rates of heat transmission, from Lang's evaporator experiments.

point *b* appear to lie on the natural direction of the curve, but for the lowest four or five curves it would seem that the points *a* do so, whilst for the highest two or three curves neither *a* nor *b* seem to be natural points on the curves. It may be presumed that the differences of temperature between the respective points *a* and *b* in Fig. 110, shown also by the curve CD in Fig. 111, are partly due to the differences between the boiling temperature of salt water and the saturation temperature over the water, and to the small elevation of temperature due to hydrostatic head.

The rate of heat transmission based upon the difference of temperature between the heating steam and the vapour is shown plotted in Fig. 112. It is seen that a gradual rise of this rate occurs up to maximum values as the temperature

difference increases, and this rise is no doubt largely due to the increased circulation of the salt water and to the increase in the flow of steam through the tubes as the evaporation increases. The attainment of maximum values is possibly due to the difficulty of keeping the tubes always wet at high rates of evaporation. Incidentally it may be mentioned that the limiting volume of vapour formed quite free from particles of salt water was practically the same at all vapour pressures within the range of the experiments. It follows that the corresponding weights of evaporation would be proportional to the vapour density.

The following particulars of experiments on a small vacuum pan have been kindly supplied to the author by T. H. P. Heriot, F.C.S., lecturer in sugar manufacture at The Royal Technical College, Glasgow. These experiments were made by Mr. Heriot, assisted by some of his students, in the Technical Chemistry department of the above-mentioned college. The vacuum pan has two copper coils made into conical spirals, one situated above the other. The length of the coils were obtained by a tape measure and the mean internal diameter estimated from the water capacity, giving the following coil dimensions :—

	Length.	Estimated internal diameter inches.	Outside dia., inches.	Heating surface inside coil, sq. ft.		Capacity of pan up to top coil.
Top coil	10 ft. 2 in.	1·182	1·31	3·486	6·93 total.	3·2 cub. ft., or, ·462 cub. ft. per sq. ft. H.S.
Bottom coil	9 ft. 9 in.	1·229	1·35	3·446		

The steam used was measured by weighing the water of condensation leaving the coils after discharging through a trap. The steam was taken to be dry saturated at the inlet to the coils, there being a separator on the covered steam pipe a few feet from the pan. The steam leaving the pan was also taken to be dry-saturated and was condensed in a surface condenser ; the evaporation was measured by weighing the discharge from the air-pump. The temperatures were measured by mercury thermometers inserted into pockets. The results obtained from the experiments are given in Table 27.

Reference to column 14 shows that the external loss of heat is quite consistent, except for Test 1, and this indicates that the experiments were carefully carried out. The values in column 18 were obtained by dividing the heat required for evaporation and heating of the feed (column 13) by the heating surface and by the values in column 9. Referring first to the tests 3 to 7, where the lower coil only was in action and evaporating water, it would be seen that the hot feed in Test 4 caused a decided increase in the rate of evaporation (column 17) and in the rate of heat transmission (column 18). This would be due partly to the influence of the hotter outlet temperature of the water of condensation from the coil and partly to the improved rate of circulation of the water. The higher steam temperatures in Tests 6 and 7 also increased the rate of evaporation and the rate of heat transmission, probably due to the higher rate of circulation of the liquid. Comparing Tests 4, 6, and 7 with Test 8 shows how the syrup causes a decided lowering of the values in columns 17 and 18, as is also shown by Tests 1 and 2. This was probably mostly due to the lower rate of circulation when syrup was being treated. The rates of heat transmission are also shown plotted in Fig. 136, p. 289, on a density of solution base, with the difference of temperature between the heating steam and the boiling liquid given opposite each point.

Some evaporation experiments have been made by H. Claassen.* The apparatus consisted of an enclosed iron cylinder, 480 mm. (18·9 in.) diameter and 1150 mm. (45·3 in.) high, containing the liquid to be evaporated. A copper heating coil, 45 mm. (1·77 in.) outside diameter and ·5 sq. metre (5·38 sq. ft.) heating surface, was placed as near as possible to the bottom of the cylinder. This apparatus stood on a weighing machine so that the weight could be ascertained at any time, flexible connections being used for the steam supply to the coil and also for the connection of the vessel to the condenser and air-pump. Temperatures were obtained with mercury thermometers and pressures by mercury columns.

The more important results are shown graphically in

* "Die Warmeübertragung bei der Verdampfung von Wasser und von wässrigen Lösungen," *Zeit. d. Ver. Deut. Ing.*, p. 418, Vol. XLVI, 1902.

TABLE 27

No. of Test	Duration of Test, hours.	Heating steam temps. °F. Inlet. Top coil.	Inlet. Bottom coil.	Outlet. Top coil.	Outlet. Bottom coil.	Feed temp. to pan. °F.	Boiling temp. in pan. °F.	Initial temp. difference. (Inlet steam temp. — Boiling temp. in pan.) °F.	Heating steam condensed in coil, lbs. per sec.	Heat supplied per sec. B.Th.U.	Evaporation lbs. per sec.
(1)	1·083	222	222	200	204	48·2	120	102	·0752	74	·0682
(2)	1·5	221·5	220	202	167	47·3	125	96	·0834	83·5	·0728
(3)	1·5	—	213	—	147	48·2	126	87	·0446	46·2	·0391
(4)	1·5	—	214	—	177	116	127·5	86·5	·0726	73·1	·0675
(5)	2	—	213	—	156	44·6	117·4	95·6	·0647	66·5	·0573
(6)	2	—	227	—	210	50	130·5	96·5	·0955	93·4	·0812
(7)	1·67	—	231	—	206	41	129·5	101·5	·0866	85·2	·0725
(8)	1·5	—	240	—	213	124·5	129·6	110·4	·0567	55·6	·0502
1	2	3	4	5	6	7	8	9	10	11	12

TABLE 27—*continued*

Heat required per sec. B.Th.U.	Heat lost per sec.		Efficiency of pan.	Evaporation per sq. ft. H.S. per hour, lbs.	B.Th.U. per sq. ft. per sec. per °F. initial temp. diff.	Remarks.
	B.Th.U.	Per cent of heat supplied.				
74·8	− ·8	—	—	35·4	·106	Syrup at 51° Brix. Level about 6½ in. above top coil and kept level by feeding in water.
80·0	+3·5	4·2	95·8	37·8	·12	Syrup at 51° Brix. Level about ¾ in. above top coil and kept level by feeding in water.
43·0	+3·2	6·9	93·1	40·8	·143	Lower coil only in action evaporating water. Level about ½ in. above top coil.
69·7	+3·4	4·65	95·4	70·5	·234	Lower coil only in action evaporating water. Level about ¾ in. above top coil.
63·0	+3·5	5·3	94·7	59·8	·191	Lower coil only in action evaporating water. Level about ·4 in. below top coil.
89·2	+4·2	4·5	95·5	84·8	·268	Lower coil only in action evaporating water. Level about 1 in. below top coil.
80·3	+4·9	5·75	94·25	75·6	·229	Lower coil only in action evaporating water. Level about top coil.
51·5	+4·1	7·4	92·6	52·5	·135	Lower coil only in action. Syrup at 56° Brix. Level about 4½ in. above top coil and kept level by feeding water.
13	14	15	16	17	18	19

Figs. 113 to 118. When evaporating water at 100° C. the rate of heat transmission, expressed in B.Th.U. per sq. ft. per sec. per degree Fahr. difference of temperature between the heating steam and the vapour increased as shown in Fig. 113 with an increase in the temperature difference. No doubt this was mostly due to the increased rate of circulation of the boiling water with the increased heat transmission or evaporation, but probably to some extent it was also due to the increased flow of steam through the coil and the higher discharge temperature of the water of condensation.

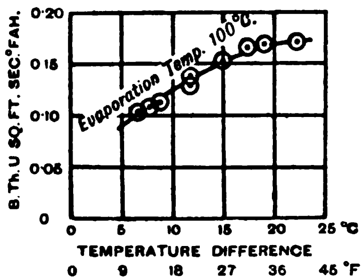

FIG. 113.—Rate of heat transmission when evaporating water, from Claassen's evaporator experiments.

Further series of experiments were then made with water at various evaporation temperatures varying from 100° C. to 60° C., altering the difference of temperature at each evaporation temperature by altering that of the heating steam. The

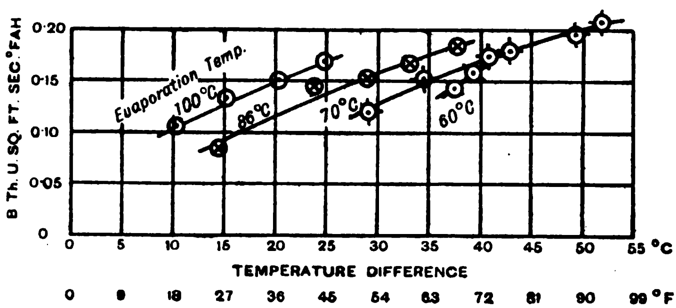

FIG. 114.—Rates of heat transmission when evaporating water, from Claassen's evaporator experiments.

results obtained are shown in Fig. 114, and it is seen that for any given temperature difference between the heating steam and the vapour the rate of heat transmission decreases with the evaporation temperature. At any constant evaporation temperature the nature of the results obtained is similar to that shown in Fig. 113.

Another series of tests was made when evaporating water with the heating steam kept constant at a series of values varying from 125° C. to 100° C., changes of the difference of temperature being obtained by altering the pressure and temperature of evaporation in each of the series. These

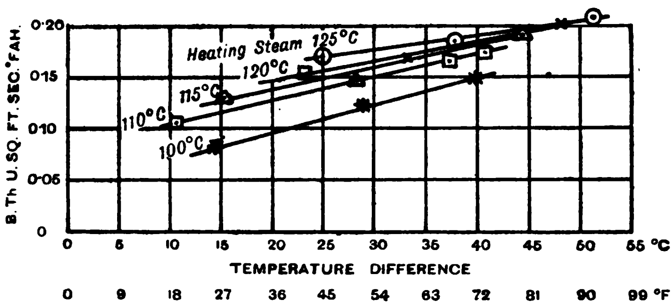

FIG. 115.—Rates of heat transmission when evaporating water, from Claassen's evaporator experiments.

results are shown in Fig. 115, and again show the general increase in the rate of heat transmission with an increase of the temperature difference. Also for any given temperature difference the lower the temperature of the heating steam the lower does the rate of heat transmission become.

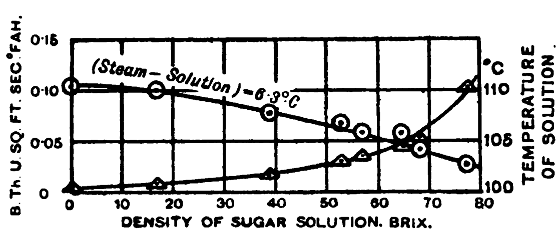

FIG. 116.—Rate of heat transmission and boiling temperature at different densities of solution of sugar, from Claassen's experiments.

Some further experiments were made with solutions of sugar, molasses, and common salt at various densities. In these the evaporation took place at practically atmospheric pressure, the boiling temperature of the solution, of course, increasing with the density. The steam temperature was arranged so as

to keep practically a constant difference of temperature between the steam and the solution. Fig. 116 refers to the tests with sugar solution with a difference between the heating steam and the solution of about 6·3° C. Under these conditions

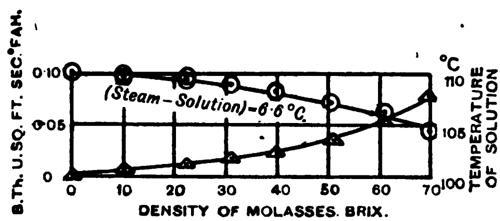

FIG. 117.—Rate of heat transmission and boiling temperature at different densities of solution of molasses, from Claassen's evaporator experiments.

it is seen that the rate of heat transmission, in B.Th.U. per sq. ft. per sec. per degree Fahr. difference between heating steam and solution, decreases as the density increases, probably because of the decreased rate of circulation of the boiling solution with the increasing density and viscosity. The boiling temperature of the solution is also shown plotted on the

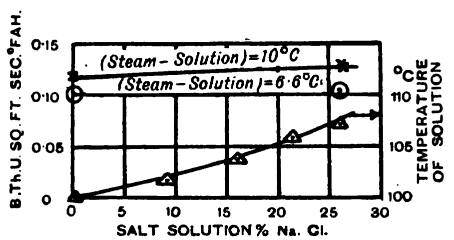

FIG. 118.—Rate of heat transmission and boiling temperature of solution of common salt, from Claassen's evaporator experiments.

density base in Fig. 116. Fig. 117 shows that the solution of molasses gave much the same results as the sugar solution. With the solution of common salt two sets of tests were made, one with a difference of temperature between the heating steam and the solution of about 6·6° C., the other at about 10° C. It is seen from Fig. 118 that in these tests the rate of heat transmission remained practically constant as the salt solution

increased in density, probably because the small increase in the viscosity of the salt solution had little or no influence on the rate of circulation.

According to the values given in the paper, taking the viscosity of water as unity, that of the sugar solution at the density 68° Brix. was 1·54,* of the molasses solution at 69·4° Brix. 1·55, and of the salt solution having 26% NaCl. 1·15.

A few experiments were made on the influence of the hydrostatic head of the boiling water on the rate of heat transmission. With an increase of head the rate of heat transmission, calculated on the difference of temperature between the heating steam and the vapour, was found to decrease slightly in accordance with the small rise of the boiling temperature with increase of head.

Some experiments were also made by Claassen on a three-effect evaporator when concentrating sugar juice. Each effect had vertical brass tubes 1370 mm. (54 in.) length, 45 mm. (1·77 in.) diameter and 2 mm. (·079 in.) thick. In the middle of each tube system was a downcomer circulating tube of 400 mm. (15·75 in.) diameter. The heating steam was supplied to the outside of these tubes. The total heating surfaces were: In 1st effect 214 sq. m. (2300 sq. ft.); in the 2nd 236 sq. m. (2540 sq. ft.), and in the 3rd 148 sq. m. (1590 sq. ft.). The connecting steam pipes between each effect were so large that there was practically no drop of pressure through the pipes.

Some of the results obtained are given in detail in Tables 28, 29, and 30. Table 28 refers to tests made with the head of liquid there specified. It will be seen that in nearly all cases the increased head caused a reduction in the rate of heat transmission. Also, as is usual in multiple evaporators, the rate was greatest in the 1st effect and least in the 3rd. This is probably mostly due to the lower rate of circulation of the boiling liquid as its density and viscosity increase. That this was the most likely cause is seen from the results of tests on the 3rd effect with different densities, as shown by Table 30. After the surfaces had been cleaned

* This value of the viscosity would appear to be much less than is represented by Table 41, p. 295.

some tests were made with the three-effect evaporator over a week's working, with the results shown in Table 29.

The rates of heat transmission, in B.Th.U. per sq. ft. per sec. per degree Fahr. difference between the heating steam

TABLE

EXPERIMENTS ON THE INFLUENCE

Effect I. High = 1 to 1·2 metres.
Effect II. „ = 1 to 1·2 metres.
Effect III. „ = 0·8 metre.

Test number		I	
Height of Juice		Low	High
Brix.-contents in Hot Juice	I	9	8
	II	25	17
	III	60	59
Temperature of Steam	Heating steam in I °C.	108·0	110·1
	Connecting pipe I to II °C.	102·0	103·7
	„ „ II to III °C.	94·8	95·5
	„ „ from III °C.	67·9	65·7
Temperature of Boiling Juice, in	I °C.	102·5	104·2
	II °C.	95·8	96·3
	III °C.	71·9	69·6
Temperature difference (Heating Steam inlet – Hot Juice) in	I °C.	5·5	5·9
	II °C.	6·2	7·4
	III °C.	22·9	25·9
Rate of Heat Transmission, B.Th.U., sq. ft., second, degree Fahr. difference between Steam and Boiling Liquid.	I	·175	·148
	II	·125	·0965
	III	·0553	·0445
Temperature difference (Heating Steam inlet – Vapour) in	I °C.	6·0	6·4
	II °C.	7·2	8·2
	III °C.	26·9	29·8
Rate of Heat Transmission, B.Th.U., sq. ft., second, degree Fahr. difference between Heating Steam and Vapour.	I	·160	·136
	II	·108	·0871
	III	·0471	·0387

and the vapour, have been plotted in Fig. 135, p. 288, on the base of the temperature of the heating steam, from the values in Tables 28 and 29. In Fig. 136 there is also shown on the base of the density of the liquid the rate of heat trans-

mission for the same experiments, though the plotted points have been omitted. In this figure is also plotted the results obtained with different liquid densities in the 3rd effect, taken from Table 30.

28

OF HEIGHT OF FLUID

Low = 0·3 to 0·4 metre.
,, = 0·4 metre.
,, = 0·2 to 0·3 metre.

II		III		IV		V		VI	
Low	High	Low	High	Low	High	Low	High	Low	High
7 21 59	6 15 50	10 27 56	10 16 45	9 24 60	6 16 55	8 24 58	9 18 53	11 22 62	10 14 56
109·9 103·7 95·1 69·1	110·1 102·6 92·6 66·0	108·8 102·2 93·6 66·2	108·9 101·6 91·6 66·7	109·3 101·8 93·8 66·0	109·5 101·9 92·8 64·5	109·4 102·7 94·3 66·2	109·5 101·3 92·0 64·5	111·6 103·7 94·6 66·9	111·7 102·9 92·9 66·5
104·2 96·1 73·1	103·1 93·4 68·7	102·7 94·6 69·4	102·1 92·4 68·9	102·3 94·8 70·0	102·4 93·6 67·7	103·2 95·3 69·9	101·8 92·8 67·5	104·2 95·6 71·3	103·4 93·6 69·8
5·7 7·6 22·0	7·0 9·2 23·9	6·1 7·6 24·2	6·8 9·2 22·7	7·0 7·0 23·8	7·1 8·3 25·1	6·2 7·4 24·4	7·7 8·5 24·5	7·4 8·1 23·3	8·3 9·3 23·1
·177 ·108 ·0621	·122 ·0741 ·0473	·161 ·105 ·053	·124 ·073 ·049	·158 ·1305 ·0627	·14 ·0975 ·0535	·169 ·117 ·0587	·137 ·1025 ·0582	·161 ·121 ·0667	·134 ·098 ·0685
6·2 8·6 26·0	7·5 10·0 26·6	6·6 8·6 27·4	7·3 10·0 24·9	7·5 8·0 27·8	7·6 9·1 28·3	6·7 8·4 28·1	8·2 9·3 27·5	7·9 9·1 27·7	8·8 10·0 26·4
·163 ·0952 ·0525	·114 ·0683 ·0425	·149 ·0927 ·0469	·116 ·0671 ·0446	·147 ·114 ·0537	·131 ·089 ·0475	·156 ·103 ·051	·129 ·0937 ·0519	·151 ·108 ·0561	·126 ·0911 ·060

An elaborate series of tests on an experimental vacuum evaporator plant of the calandria type have been made by E. W. Kerr* in the mechanical engineering laboratory of

* "Tests upon the Transmission of Heat in Vacuum Evaporators," *Trans. Amer. Soc. Mech. Engs.*, Vol. 35, 1913, p. 731.

Louisiana State University. The general arrangement of the plant and the testing arrangements are shown in Fig. 119. It will be noted that the body of the evaporator was supported above the calandria so that the calandria could be removed at

TABLE

EXPERIMENTS OVER A WEEK'S

		Monday.
Brix.-contents of Hot Juice	I	9
	II	25
	III	60
Temperature of Steam.	Heating Steam in I °C.	108·0
	Connecting pipe I to II °C.	102·0
	,, ,, II to III °C.	94·8
	,, ,, from III °C.	67·9
Temperature of Boiling Juice, in	I °C.	102·5
	II °C.	95·8
	III °C.	71·9
Temperature difference (Heating Steam inlet – Juice).	I °C.	5·5
	II °C.	6·2
	III °C	22·9
Rate of Heat Transmission, B.Th.U. sec., sq. ft., degree Fahr. difference (Steam – Juice).	I	175
	II	·125
	III	·0553
Temperature difference (Heating Steam inlet – Vapour).	I °C.	6·0
	II °C.	7·2
	III °C.	26·9
Rate of Heat Transmission, B.Th.U., sq. ft., second, degree Fahr. difference between Heating Steam and Vapour.	I	·160
	II	·108
	III	·0471

the flange A and another one bolted on. Thermometers were placed as shown in this figure to measure the temperatures in the steam pipe, in the vapour space, in the bottom of the steam compartment, in the top of the steam compartment, the entering juice or water, the water of condensation, and that in the room. These thermometers were placed somewhat

differently in the different modifications of calandria which were used. Vacuum and pressure gauges were connected to the positions shown by Fig. 119. A majority of the tests were made with water as the liquid to be boiled, the remainder

29

CONTINUOUS OPERATION

Week XIII.				Week XIV.
Tuesday.	Thursday.	Friday.	Saturday.	Sunday.
7 22 57	7 21 59	8 21 58	7 25 53	9 22 62
108·7 102·3 94·6 67·3	109·9 103·7 95·1 69·1	109·7 104·2 95·1 66·7	110·6 104·6 95·7 67·0	112·3 105·9 97·3 68·7
102·8 95·6 70·8	104·2 96·1 73·1	104·7 96·1 70·2	105·1 96·7 70·0	106·4 98·3 73·2
5·9 6·7 23·8	5·7 7·6 22·0	5·0 8·1 24·9	5·5 7·9 25·7	5·9 7·6 24·1
·181 ·128 ·0593	·177 ·108 ·0621	·203 ·1025 ·053	·178 ·0998 ·0513	·209 ·145 ·0719
6·4 7·7 27·3	6·2 8·6 26·0	5·5 9·1 28·4	6·0 8·9 28·7	6·4 8·6 28·6
·167 ·111 ·0517	·163 ·0955 ·0526	·185 ·0913 ·0465	·163 ·0887 ·046	·193 ·128 ·0605

being made with sugar solutions produced by mixing white sugar with water in such proportions as would give the desired densities. Liquor made in this manner was clean and pure and gave practically no fouling of the heating tubes. All tests were made with practically clean tubes.

In commercial evaporators it is customary to remove the

TABLE 30

EXPERIMENTS ON THE INFLUENCE OF CONCENTRATION OF JUICE IN EFFECT III

Test number	I				II			III	IV	V	VI
	1	2	3	4	1	2	3				
Brix.-contents of Hot Juice in III	0 water	20	30	65 to 70	19	30	62	61	65 to 70	54	55
Temperature of Vapour in II °C.	90·2	90·0	90·7	96·5	86·5	88·9	93·1	95·8	93·8	93·1	96·4
,, ,, ,, ,, III °C.	71·6	69·0	64·5	62·3	64·6	65·1	64·0	68·0	65·7	66·5	66·4
,, ,, Boiling Juice in III °C.	71·6	69·8	65·5	68·8	65·6	66·6	68·4	72·2	72·2	69·5	69·6
Temperature difference (Vapour in II – Juice in III) °C.	18·6	20·2	25·2	27·7	20·9	22·3	24·7	23·6	21·6	23·6	26·8
Rate of Heat Transmission (B.Th.U., sec., sq. ft., degree Fahr. difference (Steam – Juice))	·114	·106	·0895	·0605	·0895	·0878	·0685	·049	·064	·065	·0588
Temperature difference (Vapour in II – Vapour in III) °C.	18·6	21·0	26·2	34·2	21·9	23·8	29·1	27·8	28·1	26·6	30·0
Rate of Heat Transmission (B.Th.U., sq. ft., sec., degree Fahr. difference between Heating Steam and Vapour)	·114	·102	·086	·049	·0853	·0825	·0582	·0416	·0492	·0576	·0525

concentrated liquor from the last body by means of a pump. In this experimental plant, however, a slightly different arrangement was adopted. In starting a test, sugar juice of the desired density and quantity was placed in the evaporator

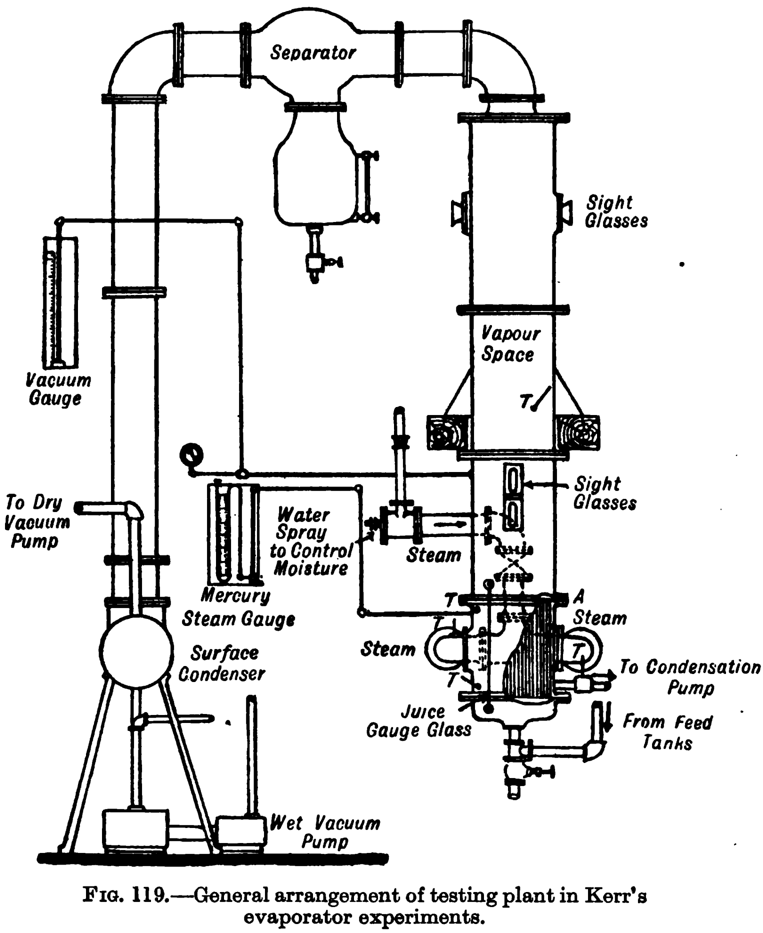

FIG. 119.—General arrangement of testing plant in Kerr's evaporator experiments.

and as the evaporation progressed water was supplied in such quantities as would keep a constant level in the juice compartment as indicated by the gauge glass. After being weighed in calibrated vessels the water was fed by gravity aided by the vacuum in the boiling vessel, a valve in the feed pipe being

evaporator above the top
hich may be called the
ll the calandrias tested
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was also provided in the

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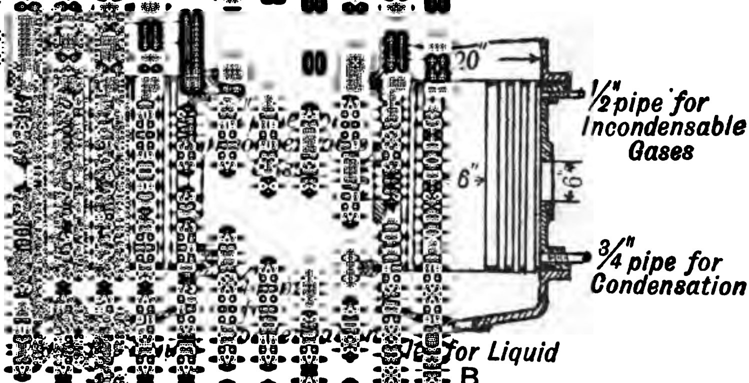
am of varying quality
r into the steam between



rr's evaporator experiments.

a. In starting a test the
and a charge of juice
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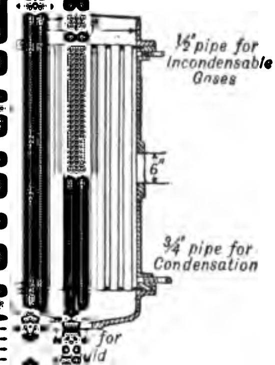

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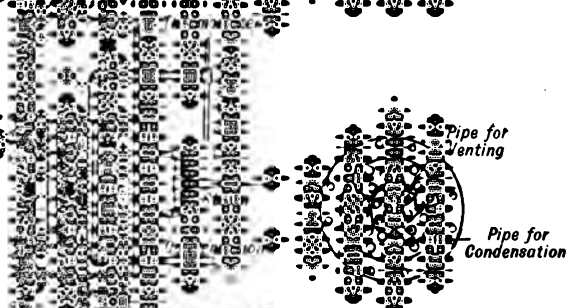


...experiments.
...us path towards
...as drawn off by
... top tube plate
... The object of
...city among the

of the air at a definite
rator shown in Fig. 123
type with special arrange-
are two thick tube plates,
ron. The heating tubes,
expanded into the upper
end. The heating steam
the air was removed by
the lower tube plate and
to near the top of the
ibes. The boiling liquid
the 2 in. heating tubes.
rostatic head of the juice,
depth of submergence of
surface, is of importance
evaporators. It is evi-
the boiling pressure in the
the corresponding satura-
perature would be higher
vapour pressure and satu-
perature above the liquid.
ple, suppose the saturation
res of the heating steam
vapour formed are 228° F.
F. respectively, giving a
re difference of 26° F. If
the juice were, say, 4 feet
e total boiling pressure at
r than the vapour pressure
er ; and the corresponding
say, 208·5° F., giving a
8—208·5=19·5° F. at the
the average difference of

., say. A similar calcula-

oiling pressure the greater
e with a given depth of
that this influence would
ltiple evaporator.

Tests were made on calandrias A, C, D and E with various degrees of submergence, under practically the same conditions of steam and vapour pressures. Water was used for boiling in all the tests. The average conditions are given in Table 31.

TABLE 31.

Type of calandria.	No. of tests in series.	Average values of absolute pressures, inches mercury.		Average values of temperatures, °F.			Water evaporated per sq. ft. of heating surface, lbs. per hour.		Height of boiling liquid above lower tube plate, inches.		Estimated average difference of temperature °F.
		Vapour.	Heating steam.	Vapour.	Heating steam.	Entering water.	Max.	Min.	Max.	Min.	
Fig. 120, A	11	26·8	33·65	206·6	218·2	82	8·14	1·32	24	3	11·2
Fig. 121, C	9	21·9	29.92	196·7	211·2	83	12·70	4·14	54	6	14·5
Fig. 123, D	8	21·6	29·80	196·4	212	70	15·92	11·80	48	6	15·25
Fig. 122, E	9	22·8	34·20	198·7	222·1	79	32·25	17·17	24	2	19·8

The rates of heat transmission obtained are shown plotted in Fig. 124 on the base of height of boiling liquid above the lower tube plate, as measured by the level in the gauge glass. At very low heads only the lower portions of the tubes were kept wet by the boiling water and therefore more or less of the heating surface was then ineffective, which accounts partly for the low rates of heat transmission based upon the total surface. This rate, however, in each case soon rises to a maximum as the head increases, these maxima probably coinciding with the conditions at which ebullition was sufficient to project the boiling water just a little over the top of the tubes. Further increase of head showed a decrease in the rate of heat transmission. It will also be observed from Fig. 124 that after the maximum is reached the rate of decrease in the rate of heat transmission as the depth of submergence increases is about the same for calandrias A and E, which had tubes 24 in. long. Similarly, the rate of decrease for calandrias C and D is about the same, the tube lengths being 48 and 54 in. respectively, but this rate of decrease is apparently less than for the calandrias A and E, showing that the influence of a definite small change of hydrostatic pressure is less for long than for short tubes.

At first sight it might appear that the greater rate of heat transmission given by calandria E, compared with calandria A, is due to the special arrangements made for the removal of air and the distribution of the flow of steam in E. Although some improvement will no doubt be due to these circumstances, an inspection of Table 31, p. 275, shows that the difference of

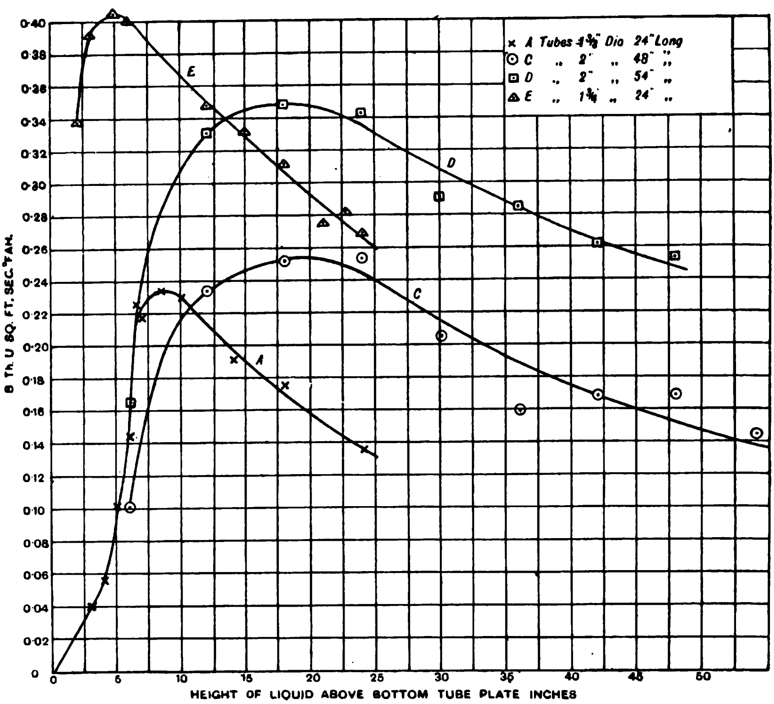


FIG. 124.—Variation of rate of heat transmission with height of water in calandria, from Kerr's evaporator experiments.

temperature during the tests was nearly twice as great in E as in A, with the result that the heat transmitted would be thereby increased and the rate of circulation of the boiling liquid correspondingly increased. This would naturally of itself tend to increase the rate of heat transmission. It follows, therefore, that with evaporators of this type, to estimate any influence on the side of the heating steam, a comparison should only be made when the steam temperatures and the heat trans-

mitted per square foot of surface were approximately the same, for then the rates of circulation of the boiling liquid would probably be somewhat similar.

Comparing the results in Fig. 124 for the calandrias D and C, and referring to Table 31, p. 275, it will be seen that the

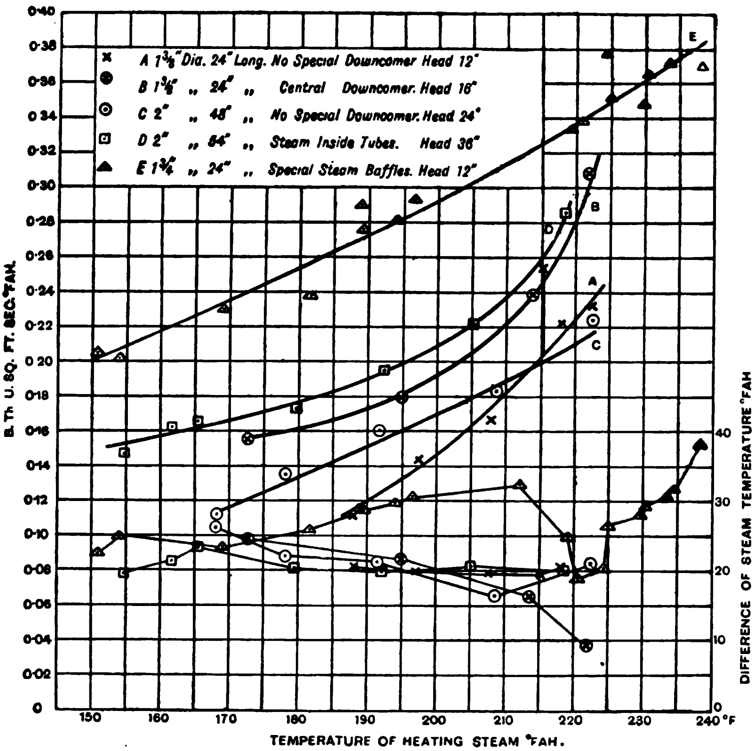


FIG. 125.—Rates of heat transmission, from Kerr's evaporator experiments.

temperatures, temperature difference, and water evaporated per square foot per hour were not much different in these two cases. There is a difference, however, in the type of evaporator, calandria D having the heating steam inside the tubes, whilst in C the heating steam was outside the tubes. The rate of circulation of the boiling liquid may consequently have been different in these two cases.

Another series of tests were conducted to ascertain the influence of the steam temperatures upon the rate of heat transmission when evaporating pure water. The extreme values quoted in Table 32 illustrate the conditions of operation, and it is seen that while the temperature of the heating steam was varied, that of the boiling water and the vapour were allowed to alter at the same time. The rate of circulation of the boiling water would no doubt be altered by these various conditions of working and would influence the rate of heat transmission accordingly. However, the rate of heat transmission for each calandria tested has been plotted in Fig. 125 on the base of heating steam temperatures, and these indicate a rapid decrease as the steam temperature decreased ; but the other conditions involved should be noted before any definite conclusion is drawn as to the influence of steam temperature. Thus, in Fig. 125 the difference of temperature between the heating steam and the vapour over the boiling water is shown plotted and indicates quite considerable variations. It would seem, however, that notwithstanding the several variable factors introduced between the different tests, calandrias A and C, which had no special downcomers, were inferior to calandria B having a central downcomer, whilst the calandrias E and D, both of which had special arrangements on the steam side for distributing the steam-flow and for extracting the air, seem to be superior to all the others.

For the purpose of studying the influence of air in the heating steam a series of tests were made on calandria A, Fig. 120, p. 272. The amount of air present in the heating steam was regulated by varying the speed of the air-pump and by admitting air into the heating steam through a pet cock. The quantity of air present was determined by the temperature method, that is, by the actual observation of the temperatures in the steam compartment the corresponding partial steam pressure was obtained from steam tables, and this subtracted from the total pressure gave the partial pressure of the air in the steam space. The location of the thermometers is indicated in Fig. 119, p. 271. Table 33 shows the extreme conditions of operation.

TABLE 32

Type of calandria.	No. of tests in series.	Water level above bottom tube plate, inches.	Absolute pressures, inches, mercury. Vapour over boiling liquid.	Absolute pressures, inches, mercury. Heating steam in calandria.	Average temperatures, °F. Heating steam in calandria.	Average temperatures, °F. Estimated temperature difference.	Average temperatures, °F. Entering water.	Water evaporated per sq. ft. heating surface, lbs. per hour.	B.Th.U. per sq. ft. per sec. per deg. Fah. difference.
A	6	12	23·72 to 12·11	35·93 19·14	222·5 188·1	21·27 20·45	82 82	16·23 7·55	·232 to ·111
B	4	16	29·84 to 8·09	35·93 14·02	222·0 172·8	9·2 24·3	83 81	9·28 12·6	·308 to ·155
C	5	24	23·88 to 7·0	36·04 12·48	222·5 168·1	21·0 26·0	82 81·5	15·44 9·75	·224 to ·112
D	7	36	22·14 to 5·3	33·72 8·38	218·5 154·8	20·0 19·4	76·1 76·0	18·9 9·5	·286 to ·146
E	18	12	22·32 to 4·1	47·65 7·6	238·3 150·9	38·2 22·44	78·0 79·0	46·25 15·45	·370 to ·205

TABLE 33

INFLUENCE OF AIR IN HEATING STEAM.

Type of calandria.	No. of tests.	Water level above bottom tube plate, inches.	Absolute pressures, inches mercury. Vapour over boiling liquid.	Absolute pressures, inches mercury. Heating steam in calandria.	Temperatures, °Fah. Vapour over boiling liquid.	Temperatures, °Fah. Heating steam in calandria.	Temperatures, °Fah. Entering water.	Estimated temperature difference, °F.	Steam temperature above or below saturation corresponding to total pressure.	Water evaporated per sq. ft. heating surface lbs. per hour.	$\frac{P_s}{P_t}$	B.Th.U. per sq. ft. per sec. per °F. difference.
A	16	12	8·06 to 8·04	17·64 17·57	151·8 152·0	186·6 168·3	80 80	34·8 15·8	+0·1 −18·0	19·1 4·61	1·0 ·668	·165 to ·041

The rate of heat transmission is shown plotted in Fig. 126 on the base $\frac{P_s}{P_t}$, and it will be observed how rapidly this rate decreases as $\frac{P_s}{P_t}$ decreases, that is, as the partial air pressure in the heating steam space increases. On referring to Table 33 it is seen that, while the total pressures were kept practically constant in this series, the partial pressure of the air reduced considerably the temperature of the heating steam. In the first test, although the ratio $\frac{P_s}{P_t}$ is given as unity, it is hardly likely that all air was excluded; the method of estimating the partial air pressure from measurement of the steam temperature is not very accurate when only a small amount of air is present. The decrease in the rate of heat transmission shown in Fig. 126 may not all be due to the increase of resistance to heat-flow on the heating steam side of the tubes; for it is seen from Table 33 that the water evaporated per square foot per hour varied from 19·1 to 4·61 lb., and, of course, the rate of circulation of the boiling water varied accordingly. This of itself would cause an increase of resistance on the water side of the tubes and would reduce the rate of heat transmission accordingly.

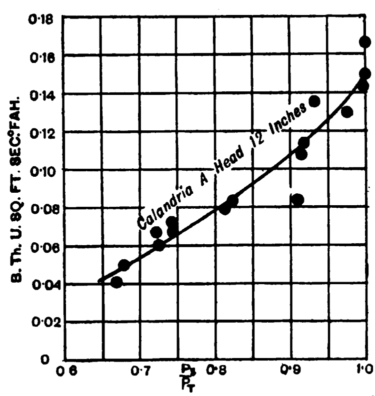


FIG. 126.—Relation between rate of heat transmission and ratio of steam pressure to total pressure, from Kerr's evaporator experiments.

A series of experiments were made upon calandria E with densities of sugar juice varying from 18 to 70 Brix., the height of the juice, as shown by the gauge glass, being about 12 inches above the lower tube plate. By subtracting the saturation temperature of the vapour from the average observed temperature of the juice the increase in the boiling temperature due to the combined effect of juice density and hydrostatic head

was obtained. The results are shown plotted in Fig. 127 on a density (Brix.) base. Where the curve cuts the line of zero density indicates approximately the rise of boiling temperature due to the hydrostatic head.

Another set of experiments was made to determine the

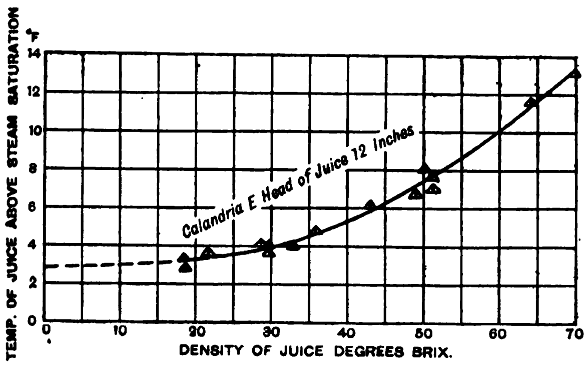


FIG. 127.—Temperature of sugar-juice above saturation temperature of vapour, from Kerr's evaporator experiments.

variation of the rate of heat transmission with density of the juice, and the results obtained are plotted in Fig. 128 on the density base. Table 34 also illustrates the extreme range of values obtained.

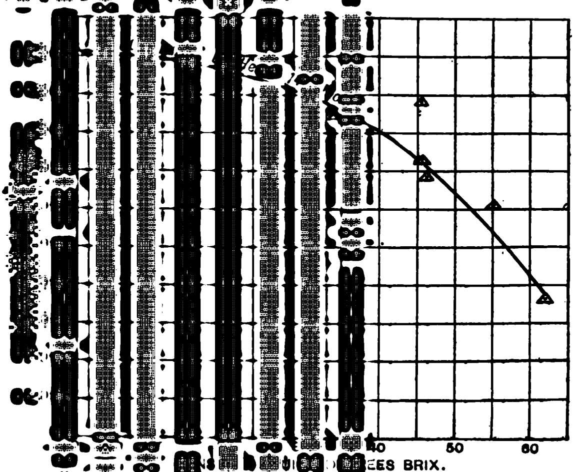
TABLE 34

TESTS WITH VARYING DENSITY OF BOILING LIQUID (HEAD 14 INCHES).

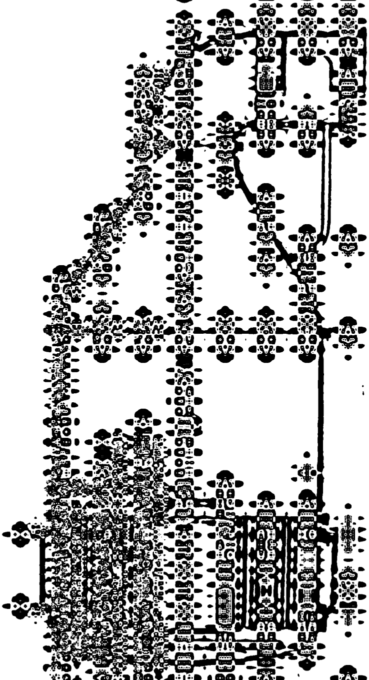
Type of calandria.	No. of tests in the series.	Absolute pressures inches mercury.		Temperatures, °F.			Density of juice, deg. Brix.	Water evaporated per sq. ft. per hour, lbs.	B.Th.U. per sq. ft. per sec. per deg. Fah. diff.
		Vapour over boiling liquid.	Heating steam in calandria.	Vapour over boiling liquid.	Heating steam in calandria.	Juice entering.			
E	7	5·06 to 4·65	8·60 9·53	134·8 135·1	156·9 160·6	81·0 81·0	0 61·86	4·93 3·0	·21 to ·073

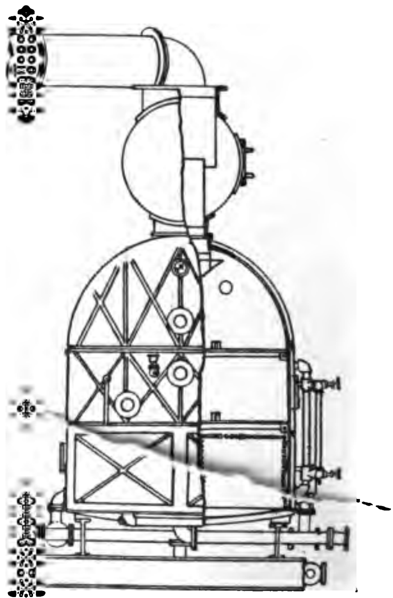
The decrease of the heat transmitted as the density of the boiling juice increased might be ascribed to the following causes :—

(1) The boiling temperature of the juice increases with the



... t transmission with ... orator experiments.





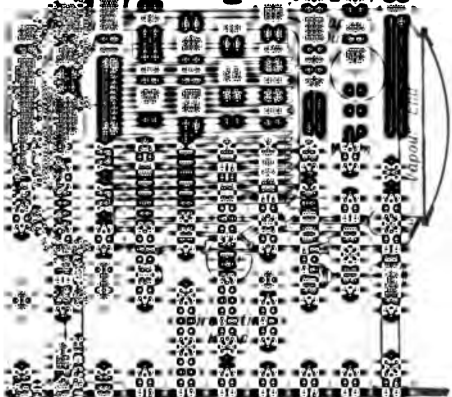
... IG. 130.—Evaporator B, horizontal tubes, tested by Kerr.

... liquid and thus ... the heating steam

... with the density ...

... uces the rate of

... hown in Fig. 128 ... re between the ... the boiling juice.



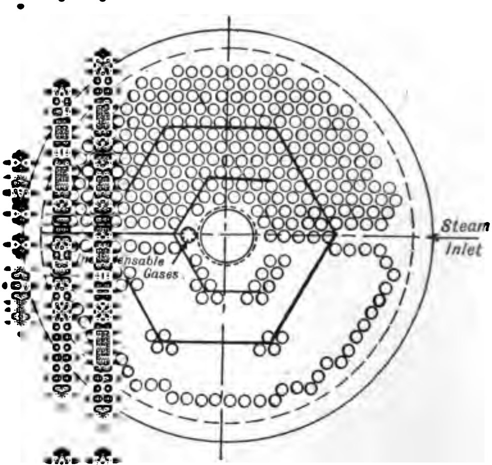
..., with ... r.

... a single-effect ... ll-sized multiple- ... ns for the manu- ... included double, ... s represented in ...

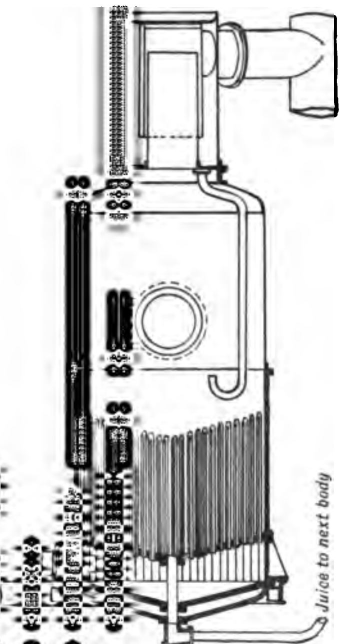
... ion, has vertical ... plied in the shell ... the air vented or ... downcomer tube. ... ere the steam is

...," *Jour. Amer. Soc.*

the inside and the juice
ese tubes. Type C (Fig.
izontal steam tubes ex-
ne end into a thick tube
ese tubes are closed at
d except for the small
which releases the incon-
s into the vapour space.
condensation leaves the
e end where the steam
is supplied by centrifugal
distributor above the
ls in a film from one tube
The piping is arranged so
ce may be recirculated
es until the required con-
each body is obtained.
. 132) has vertical tubes
ly 20 feet long, with the
and the steam outside
The level of the juice is
tom of the tubes and the



—Evaporator E, similar to A,
9, but with baffles in steam space,
by Kerr.



...vaporator G, tested ...y Kerr.

...x.

... B(W−X),

... 100.

...tage evaporation.

...as also calculated

TABLE 35

No. of test.	Duration hours.	Type of Evaporator.	No. of effects.	Length of tubes, ft.	Dia. of tubes, in.	Heating surface of each body, sq. ft.	Steam pressure in 1st effect, lbs. sq. in. absolute.	Steam pressure in last effect. lbs. sq. in. absolute.	Temperatures, °F.		Density of juice in deg. Brix (at 17·5° C.	
									Juice entering.	Juice leaving.	Entering.	Leaving.
1	7·13	A	4	4½	2	4505	16·3	1·8	183·5	101	13·7	50·2
7	4·92	A	2	4	2	1901	14·9	5·2	206·7	128·9	15·9	56·0
15	4·0	B	4	9½	¾	2000	21·4	3·8	199·7	129·2	17·5	62·9
17	6·0	B	4	13	⅞	3432	18·7	3·7	181	—	13·0	57·5
23	6·0	C	4	7¼	4½	2570	19·1	5·9	194	141	12·7	53·2
28	6·03	D	4	—	—	4000	17·9	4·9	207·7	136·9	12·6	46·2
30	4·0	D	3	23	—	2112	14·8	2·3	212·2	125	16·5	46·2
33	6·0	E	3	4	2	3054	14·6	4·6	171	136	14·9	58·0
38	5·69	G	4	4	2	1529	15·4	5·8	188·1	147·4	15·0	64·2

TABLE 35—*continued*

Method of removing water of condensation. *	Method of venting. *	Percentage evaporation.		Water evaporated, lbs.	Water evaporated per pound steam supplied, lbs.	Water evaporated per sq. ft. H.S. per hr., lbs.	Radiating surface of evaporators per 100,000 gallons of juice per 24 hours, sq. ft.	Percentage of radiating surface covered.	Heat efficiency, per cent.	No. of days since last cleaning.
		By weight.	By volume.							
f	*bc*	72·7	76·6	587,914	—	5·44	—	—	—	6
e	*ad*	71·6	76·1	104,775	2·09	5·67	—	—	94·5	—
Note 2.	*d*	72·2	77·2	248,701	3·70	7·77	1207	73·5	89·1	3½
e	*c*	77·5	81·4	314,062	3·4	5·08	1780	0	90·5	0
g	—	76·1	79·9	556,923	3·71	9·02	832	100	96·9	0
f	*Note* 3.	72·7	76·7	748,958	4·24	7·76	774	95	94·8	2
f	None.	64·3	68·5	191,585	2·91	7·77	946	0	86·5	—
f	Fig. 133.	75·4	78·8	338,859	2·49	6·27	990	40	97·4	1
e	Fig. 134 (bottom).	76·7	81·2	206,917	—	5·95	—	—	—	3

* *a.* Through shell at side; *b.* Through top tube sheet; *c.* Direct to condenser; *d.* Body to body; *e.* Pumps for each body; *f.* Siphon, body to body; *g.* Barometric leg pipe. *Note* 2. Centrifugal pumps, each body; *Note* 3. 2-in. vents at side, bottom and top, body to body.

Then, $W = va_1\rho$.

and, $W - X = Va_2\rho = va_1\rho - X$.

From previous calculation, $X = \frac{B-b}{B} W = \frac{B-b}{B} va_1\rho$

Thus, $Va_2 = va_1 \frac{b}{B}$

But, Percentage evaporation by volume $= \frac{v-V}{v} \times 100$.

$$= \left(1 - \frac{a_1 b}{a_2 B}\right) 100.$$

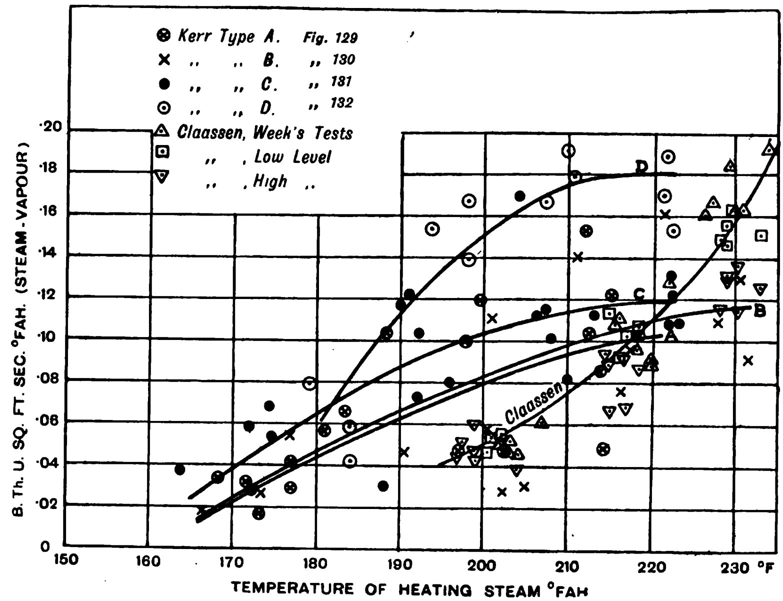


FIG. 135.—General relation between rate of heat transmission and temperature of heating steam in evaporators.

The rates of heat transmission obtained from some of the types experimented upon are shown plotted in Fig. 135 on the base of temperature of the heating steam, and the mean lines drawn in are only intended to show the general relations. Comparing these values with those obtained from the single-effect evaporator when evaporating water, shown in Fig. 125, p. 277, it would be seen that although the multiple-effect

evaporator results were somewhat erratic, they are distinctly lower than those for water in Fig. 125, as would be expected from the influence of the density of the solution on the boiling temperature, and of the viscosity on the rate of circulation. For purposes of comparison the results obtained from Claassen's experiments, given in Tables 28 and 29, p. 266, are also plotted in Fig. 135. On the whole the proportionate decrease

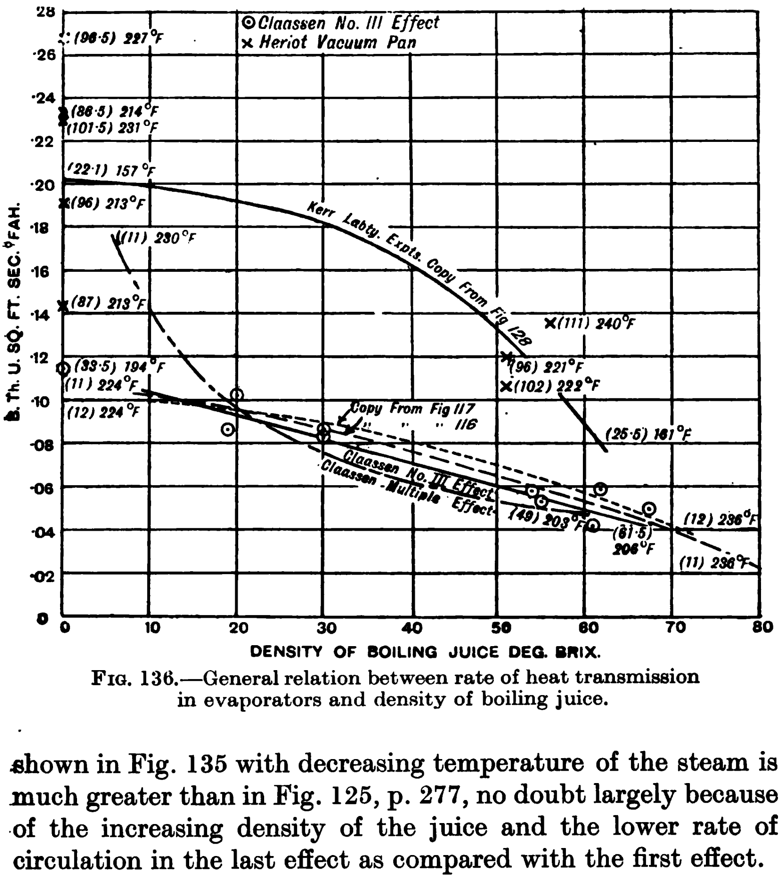


FIG. 136.—General relation between rate of heat transmission in evaporators and density of boiling juice.

shown in Fig. 135 with decreasing temperature of the steam is much greater than in Fig. 125, p. 277, no doubt largely because of the increasing density of the juice and the lower rate of circulation in the last effect as compared with the first effect.

The whole of the results available with varying density of the sugar juice have been collected together in Fig. 136, and the difference of temperature between the steam and the vapour or the fluid is marked (in brackets) against the various

points or at the ends of the curves, as well as the temperatures of the heating steam. The plotted points for the multiple-effect experiments made by Claassen have been omitted. The experiments by Kerr on multiple-effect apparatus are not shown here because the density of the juice in the intermediate effects was not given in his paper, but comparing values for the first and last effects, corresponding to the values near the two extremes of density in Fig. 136, it would be seen that the average values from Kerr's tests lie near to the curve representing Claassen's multiple-effect results.

TABLES AND DATA

TABLE 36

THERMAL PROPERTIES OF WATER AT SATURATION PRESSURE*

(From Marks and Davis' Steam Tables)

Temperature, °F.	Saturation Pressure, lbs. sq. in. absolute.	Density, lbs. per cu. ft.	Specific Heat.
20	0·06	62·37	1·0168
40	0·12	62·43	1·0045
60	0·26	62·37	0·9990
80	0·51	62·22	0·9970
100	0·95	62·00	0·9967
120	1·69	61·71	0·9974
140	2·89	61·38	0·9986
160	4·74	61·00	1·0002
180	7·51	60·58	1·0019
200	11·52	60·12	1·0039
220	17·19	59·63	1·007
240	24·97	59·11	1·012
260	35·42	58·55	1·018
280	49·18	57·96	1·023
300	67·00	57·33	1·029
320	89·63	56·66	1·035
340	118·0	55·94	1·041
360	153	55·18	1·048
380	196	54·36	1·056
400	247	53·5	1·064
420	308	52·6	1·072
440	381	51·7	1·082
460	465	50·7	1·091
480	565	49·7	1·101
500	684	48·7	1·112
520	822	47·6	1·123
540	977	46·3	1·134
560	1152	44·9	1·146
580	1349	43·3	1·158
600	1574	41·8	1·172
620	1827	40·1	—
640	2111	38·1	—
660	2428	35·6	—
689	2947	20·5	—

* For full data respecting the pressure, temperature, volume and heat contents of saturated and superheated steam refer to steam tables by Marks and Davis, Peabody, Smith and Warren, or Callendar. The various values differ slightly as given by the different steam tables, but the two last mentioned, based on Professor Callendar's work, are now considered to be founded on the most reliable data.

TABLE 37

BOILING POINT OF SOLUTIONS (GERLACH)

Glycerol, per cent	100	95	90	80	70	60	50	40	30	20	10
Vapour pressure at 100° C. mm. mercury	64	162	247	396	496	565	618	657	690	717	740
Boiling point at 760 mm. mercury °C.	290	164	138	121	113·	109	106	104	102·8	101·8	100·9

Sodium chloride, per cent	3·4	6·6	12·4	17·2	21·5	25·5	29·5	33·5	37·5
Boiling point at 760 mm. mercury °C.	100·5	101	102	103	104	105	106	107	108

TABLE 38

GLAISHER'S FACTORS FOR WET AND DRY BULB HYGROMETER*

Dry Bulb Temperature, °F.	Factor.	Dry Bulb Temperature, °F.	Factor.
Below 24	8·5	34–35	2·8
24–25	6·9	35–40	2·5
25–26	6·5	40–45	2·2
26–27	6·1	45–50	2·1
27–28	5·6	50–55	2·0
28–29	5·1	55–60	1·9
29–30	4·6	60–65	1·8
30–31	4·1	65–70	1·8
31–32	3·7	70–75	1·7
32–33	3·3	75–80	1·7
33–34	3·0	80–85	1·6

* Temperature of dew point given by multiplying the difference between the wet and dry bulb readings by the appropriate factor and subtracting the product from the dry bulb temperature. The partial pressure of the vapour is given by steam tables at the dew-point temperature. The weight of vapour and the heat contents per pound of air can be derived from the graphs in Fig. 102.

TABLE 39

DENSITY AND SPECIFIC GRAVITY OF SUGAR SOLUTIONS

(At 17½° Cent. or 63½° Fah.)

Deg. Baumé.	Sugar per cent. or Brix.	Specific gravity.	In one gallon (Imperial.)	
			lbs. sugar.	lbs. water.
0·0	0·00	1·0000	·0000	10·0000
0·5	0·90	1·0035	·0903	9·9447
1·0	1·80	1·0070	·1812	9·8888
1·5	2·69	1·0105	·2718	9·8332
2·0	3·59	1·0141	·3640	9·7770
2·5	4·49	1·0177	·4569	9·7201
3·0	5·39	1·0213	·5504	9·6626
3·5	6·29	1·0249	·6446	9·6044
4·0	7·19	1·0286	·7395	9·5465
4·5	8·09	1·0323	·8351	9·4879
5·0	9·00	1·0360	·9324	9·4276
6·0	10·80	1·0435	1·1269	9·3081
7·0	12·61	1·0511	1·3254	9·1856
8·0	14·42	1·0588	1·5267	9·0613
9·0	16·23	1·0667	1·7312	8·9358
10·0	18·05	1·0746	1·9396	8·8064
11·0	19·87	1·0827	2·1513	8·6757
12·0	21·69	1·0909	2·3661	8·5429
13·0	23·52	1·0992	2·5853	8·4067
14·0	25·35	1·1077	2·8080	8·2690
15·0	27·19	1·1163	3·0342	8·1288
16·0	29·03	1·1250	3·2658	7·9842
17·0	30·87	1·1339	3·5003	7·8387
18·0	32·72	1·1429	3·7395	7·6895
19·0	34·58	1·1520	3·9836	7·5364
20·0	36·44	1·1613	4·2317	7·3813
21·0	38·30	1·1707	4·4837	7·2233
22·0	40·17	1·1803	4·7412	7·0618
23·0	42·05	1·1901	5·0043	6·8967

Deg. Baumé.	Sugar per cent. or Brix.	Specific gravity.	In one gallon (Imperial.)	
			lbs. sugar.	lbs. water.
24·0	43·94	1·2000	5·2728	6·7272
25·0	45·83	1·2101	5·5458	6·5552
26·0	47·73	1·2203	5·8244	6·3786
27·0	49·63	1·2308	6·1084	6·1996
28·0	51·55	1·2414	6·3994	6·0146
29·0	53·47	1·2522	6·6955	5·8265
30·0	55·47	1·2632	7·0069	5·6261
31·0	57·34	1·2743	7·3068	5·4362
32·0	59·29	1·2857	7·6229	5·2341
33·0	61·25	1·2973	7·9459	5·0271
34·0	63·22	1·3091	8·2761	4·8149
35·0	65·20	1·3211	8·6135	4·5975
36·0	67·19	1·3333	8·9584	4·3746
37·0	69·19	1·3458	9·3115	4·1465
38·0	71·20	1·3585	9·6725	3·9125
39·0	73·23	1·3714	10·0427	3·6713
40·0	75·27	1·3846	10·4218	3·4242
41·0	77·32	1·3981	10·8101	3·1709
42·0	79·39	1·4118	11·2082	2·9098
43·0	81·47	1·4267	11·6233	2·6437
44·0	83·56	1·4400	12·0326	2·3674
45·0	85·68	1·4545	12·4621	2·0829
46·0	87·81	1·4694	12·9028	1·7912
47·0	89·96	1·4845	13·3325	1·5125
48·0	92·12	1·5000	13·8180	1·1820
49·0	94·30	1·5158	14·2939	·8641
50·0	96·51	1·5319	14·7843	·5437

TABLE 40

SPECIFIC GRAVITY

SODIUM CHLORIDE SOLUTION AT 15° C. (59° F.)

Na. Cl. per cent	1	2	4	6	8	12	16	20	24	26
Specific gravity	1·0072	1·0145	1·029	1·044	1·058	1·089	1·119	1·151	1·184	1·201

GLYCEROL SOLUTION

Glycerol, per cent	100	90	80	60	40	20	10	5
Specific gravity	1·263	1·237	1·210	1·156	1·101	1·049	1·024	1·012

ALKALINE SOLUTIONS

Alkali, per cent		1	2	5	10	20	30	40	50	60	70
Specific gravity for solutions of	NH_3	·996	·991	·979	·959	·925	·898	—	—	—	—
	NaOH	1·012	1·024	1·058	1·115	1·225	1·332	1·437	1·540	1·643	1·748
	KOH	1·009	1·017	1·041	1·083	1·177	1·288	1·412	1·539	1·667	1·790

TABLE 41

COEFFICIENT OF VISCOSITY OF SOLUTIONS OF CANE SUGAR
(Based on Hosking's results).

Temperature.		*Coefficient of Viscosity μ.		Per cent cane sugar in solution.
°C.	°F.	Dynes per sq. cm.	Lbs. per sq. ft.	
0	32	·1476	·000308	40
12·79	55	·0798	·000167	
20·23	68·4	·0600	·000125	
40·44	104·7	·0310	·000065	
60·51	141	·0188	·000039	
80·32	176·4	·0128	·000027	
0	32	·0372	·000078	20
13·02	55·5	·0235	·000049	
20·04	68·1	·0191	·000040	
40·31	104·5	·0117	·000025	
60·20	140·3	·0080	·000017	
80·21	176·2	·0058	·000012	
0	32	·0244	·000051	10
10·42	50·8	·0172	·000036	
20·17	68·3	·0132	·000027	
40·30	104·5	·0084	·000017	
60·27	140·5	·0059	·000012	
80·31	176·4	·0045	·0000093	
0·10	32·2	·0204	·000042	5
12·58	54·6	·0138	·000029	
20·29	68·6	·0113	·000023	
40·40	104·6	·0072	·000015	
60·34	140·6	·0052	·000011	
80·0	176	·0040	·0000083	
0·10	32·2	·0180	·000038	1
13·32	56	·0122	·000025	
20·00	68	·0103	·000022	
40·18	104·4	·0067	·000014	
60·54	141	·0047	·0000098	
80·06	176·1	·0036	·0000075	

* Measured in dynes per square cm. when the relative velocity is 1 cm. per second between two planes 1 cm. apart (no convective motion in the fluid other than the natural diffusion of the molecules) or, measured in lbs. per sq. ft. when the relative velocity is 1 ft. per second between two planes 1 ft. apart (no convective motion other than natural diffusion).

TABLE 42

COEFFICIENT OF VISCOSITY OF SOLUTIONS OF SODIUM CHLORIDE

(Based on Hosking's results).

Temperature.		*Coefficient of Viscosity.		Per cent of salt in solution.
°C.	°F.	Dynes per sq. cm.	Lbs. per sq. ft.	
·72	33·3	·0259	·000054	20
12·48	54·4	·0185	·000039	
21·20	70·2	·0150	·000031	
40·57	105	·0101	·000021	
60·05	140·1	·0074	·000015	
80·88	177·5	·0056	·000012	
·06	32·1	·0203	·000042	10
11·80	53·2	·0145	·000030	
20·46	68·9	·0118	·000025	
40·30	104·5	·0080	·000017	
60·67	141·1	·0058	·000012	
80·27	176·5	·0045	·0000094	
0	32	·0186	·000039	5
10·89	51·6	·0136	·000028	
20·22	68·4	·0108	·000022	
40·86	105·5	·0071	·000015	
60·84	141·4	·0052	·000011	
80·25	176·4	·0040	·0000083	
0	32	·0180	·000038	1
11·76	53·2	·0125	·000026	
20·26	68·5	0101	·000021	
40·15	104·3	·0066	·000014	
60·39	140·7	·0049	·000010	
80·32	176·5	·0037	·0000076	

See note, Table 41.

TABLE 43

COEFFICIENT OF VISCOSITY OF GLYCEROL AND GLYCEROL SOLUTIONS *

(Based on Schöttner's results).

Glycerol	Temperature	°C.	2·8	7·4	13·6	20·3	26·5
		°F.	37	45·3	56·5	68·6	79·7
	Coeff. of Viscosity	Dynes per sq. cm.	42·2	26·8	14·8	8·3	4·94
		Lbs. per sq. ft.	·088	·056	·0308	·0173	·0103

Glycerol solution	Temperature	°C.	8·5	8·3	8·5	5·9	14·9	21·1	8·5	8·5
		°F.	47·3	46·9	47·3	42·6	58·8	69	47·3	47·3
	Coeff. of Viscosity	Dynes per sq. cm.	7·44	3·55	1·02	·667	·390	·280	·222	·092
		Lbs. per. sq. ft.	·0155	·0074	·0021	·0014	·00081	·00058	·00046	·00019
Glycerol percentage in solution			94·5	89·9	80·3	75	75	75	64	49·8

* See note, Table 41.

For the viscosity of water, air, etc., refer to Table 9, p. 65, and Table 8, p. 64, of *Heat Transmission by Radiation, Conduction, and Convection.*

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